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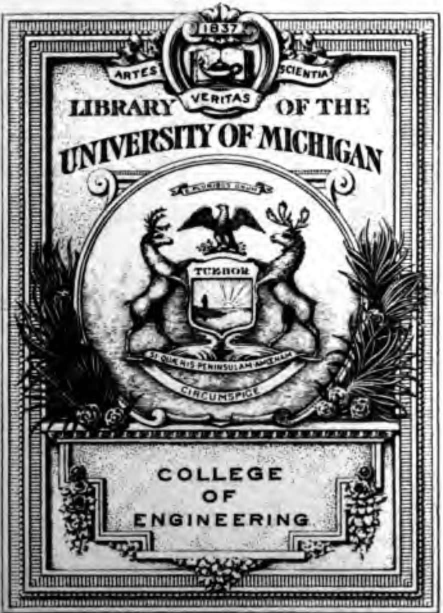

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THE GAS-ENGINE HANDBOOK

A MANUAL OF USEFUL INFORMATION

FOR

THE DESIGNER

AND

THE ENGINEER

By E. W. ROBERTS, M. E.

Author of the Automobile Pocketbook and
other works on Gas Engineering

NINTH EDITION FOURTEENTH THOUSAND

ENTIRELY REWRITTEN AND ENLARGED

THE GAS ENGINE PUBLISHING CO.
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NOTE.

It is the earnest desire of the publishers, and of the author, that this work shall continue to be the standard of reference for those interested in gas enginery. They will, therefore, be pleased to have their attention called to any errors of commission or omission in this book.

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PREFACE.

Since the advent of the first edition of the Handbook in 1900, gas-engine practice has undergone a considerable change. The automobile at that time was just fairly started as an industry and the airplane was far from realization. It was therefore imperative that instead of revising the book for this edition, it be entirely rewritten and brought up to date. The Handbook had the honor of being the first American work containing formulas for the design of gas engines, and there was little of foreign publication to draw from. For this reason the formulas were derived in many cases from standard practice of that period, being based on the dimensions of engines then in use. These formulas have all been checked on the lines of recent development and recent tables and works on design as well as notes the author has made from time to time, have been referred to for this purpose. New formulas have been added, and the discussion has been broadened both on subjects discussed in former editions and on new subjects.

The discussion of frame design has been based to a large extent on a similar discussion in the excellent work of Herr Gueldner. The proportions for the frame are based on numerous samples examined, and no such basic design has been offered before, to the author's knowledge. The discussion of the two-cycle is based on the author's personal experience in the design and the manufacture of this type.

A rather more extensive discussion of testing is given than would appear warranted in a work of this size, but

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the subject is of considerable importance, and the actual methods employed are not generally understood. This chapter has been extended and the calorimetry of gases explained. The author realizes that the plan of conducting a test, as outlined, does not agree with the recommendations of other writers and societies in the general arrangement or in the importance attached to some of the determinations. Actual practice in commercial testing will, the author believes, show the value of the plan given.

The Gas-Engine Handbook has been written as an epitome of gas-engine practice and as a handy book of reference, and not as an extended treatise. For this reason, much that the author would have liked to include in the book has been omitted for lack of space. He has therefore given such matter as he has judged most useful, and which he has had the most frequent occasion to look up himself.

E. W. ROBERTS.

SANDUSKY, O., August, 1913.

PREFACE TO NINTH EDITION.

In this edition quite a few paragraphs have been revised, and many re-written, to bring the matter strictly in accordance with the most recent practice.

E. W. ROBERTS.

CINCINNATI, O., August, 1917.

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CHAPTER I.

INTRODUCTORY.

The Gas Engine is a type of internal combustion motor, and includes all classes of prime movers operating on combustible gases or the vapors of combustible liquids, which are burned within the cylinder of the engine. It is the only internal combustion motor in use today, all heat engines not properly coming under this head being external combustion motors.

The Gas Engine therefore includes all motors operating on gasoline, kerosene (paraffine oil), the distillates and other petroleum derivatives and alcohol, wherein the combustion takes place within the cylinder of the motor.

The motors operating on the various fuels are generally named after the fuels employed, as gasoline engines, alcohol engines, etc. The terms engine and motor as used with reference to gas engines are synonymous, motor often being the term employed with reference to those operating on the liquid fuels.

The first conception of the gas engine appears to have been by one Abbe Hautefeuille, who proposed a motor driven by gunpowder as early as 1678. From that time on until 1862, many propositions were made and some partially successful engines were built by various experimenters. In the latter year there were the Lenoir and the Hugon gas engines; operative, but very wasteful of fuel.

In 1862, M. Beau de Rochas, a French engineer, made the following propositions, which were embodied in a patent of that date:

I. The largest cylinder capacity with the smallest circumferential surface.

II. Maximum speed of piston.

III. Greatest possible expansion.

IV. Highest pressure at the beginning of the stroke.

These propositions, he averred, should be the aim of every gas engine designer to carry out, in order to obtain the highest efficiency.

With the exception of proposition II, every engine since the successful event of the Otto, has been designed with these principles in view. Owing to practical limitations, extremely high piston speeds have not been found advisable, especially in stationary motors. However, since the first edition of the HANDBOOK was written, we find a tendency toward extremely high speed, especially in motors for automobiles, medium and high speed boats, and airplane motors.

It was M. Beau de Rochas who first proposed and patented what was afterward popularly known as the "Otto Cycle." It was, however, first embodied in a practical working engine by Dr. Otto. This cycle requires four strokes of the piston for the completion. The first stroke draws in a charge of gas and air; the second stroke compresses the charge, near the end of this stroke ignition takes place, followed by explosion of the gaseous mixture; expansion during the third stroke; and during the fourth stroke, the products of combustion are expelled from the cylinder.

Dr. N. A. Otto first put this cycle into practical use in 1876, when he brought out his Otto Silent engine, thereby producing a motor which so far surpassed in economy all gas engines then in use as to drive its competitors from the field. The credit of the first conception of the principles involved belongs to Beau de Rochas, although full credit should be given to Dr. Otto for embodying these principles in a practical working machine.

The cycle above described is now generally signified by calling the engine using it, a *four-cycle* engine. This is an abbreviation of four-stroke cycle, because of the four strokes of the piston required for its completion. The use of this term is now general.

The operation of the four-cycle motor can be better understood by reference to Fig. 1. Four views are shown, illustrating the operations which take place in each of the four strokes which are required to complete the cycle, and the portion of the indicator diagram produced during each stroke. In explanation of the indicator diagram, the line *A-B* is the atmospheric line and the vertical distance of any point above this line is a measure of the pressure in pounds gauge of that point. For example, the pressure at *C* is measured by the distance from the line *A-B* to *C*

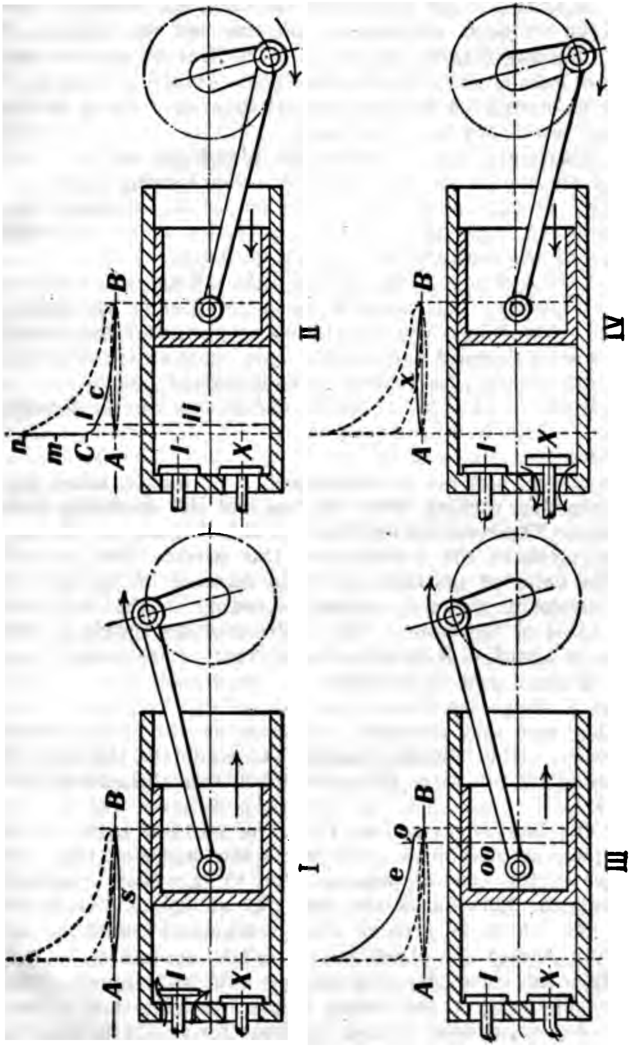

FIG. 1.

and is 85 lbs. That portion of the diagram produced during the stroke is shown as a full line and the balance of the complete diagram is dotted. The line of suction and that of exhaust have the pressure scale exaggerated so they may be distinguished from *A-B*. If they were drawn to the proper scale they would not show.

Diagram I, Fig. 1, shows the piston on the first outward or suction stroke, the inlet valve *I* being open and the air being drawn in as indicated by the arrows. The exhaust valve, *X*, is closed. That portion of the diagram given by the indicator for this stroke is the suction line *s*, which is below *A-B*, as during this stroke the pressure in the cylinder falls about 2 lbs. below that of the atmosphere. The inlet valve usually opens just after the piston has started forward and remains open until after the piston has started back, for reasons to be explained later.

Diagram II, Fig. 1, shows the piston on the second stroke, which is inward, and called the compression stroke. During this stroke both valves are closed and the mixture of fuel and air is compressed to 60 lbs. or more, the compression varying with the fuel and the operating conditions. The compression line *c* is that portion of the diagram given by the indicator for this stroke. The point *i*, is the point of ignition and is in advance of the end of the stroke a distance varying according to the fuel and the speed of the motor. The position of the piston at the time of ignition is directly below that of the point *i* on the diagram and is indicated by the dotted line *ii*. The point *C* shows the pressure at the end of the compression stroke, and this pressure is known as the compression pressure. It is slightly higher than would be the case if ignition did not start the explosion before the end of the stroke.

The line *m*, extending from the ignition point to *n*, the point of maximum pressure, is the explosion line. It indicates the rise of pressure due to the heat produced during the burning of the fuel. If an equal amount of the gas which is present after combustion could be as rapidly heated by other means, while contained in the cylinder, an equal rise of pressure would be obtained. The time required for the entire mixture to take fire is not long, but an allowance must be made for it. It is usually desired to secure the point of highest pressure *n*, at the beginning of the expansion stroke. The period of time required from the moment of ignition until the entire body of the mixture is set fire is known as the *period of inflammation.* The actual duration of this period varies greatly according to conditions and will be discussed in a later chapter.

Diagram III, Fig. 1, shows the piston on the third or expansion stroke, also called the "explosion stroke" and the "working" or "power stroke." This is the second outward stroke. During this stroke the piston moves forward with pressure, behind it, greater than the pressure given to the mixture by the piston during the compression stroke. In fact the space or area between the line *c* and the expansion line *e*, indicates the work derived from the combustion of the fuel. Note that during this stroke, as in the compression stroke, both valves are closed. In order to reduce the pressure behind the piston on the next stroke during the expulsion of the exhaust gases, it is best to open the exhaust valve early so that the pressure inside the cylinder will reach that of the atmosphere just at the end of the stroke. For this reason the exhaust valve *X*, is opened when the piston has reached the position *oo*, just below *o*, on the diagram. The sudden downward curve of the expansion line from the point *o*, to the end of the stroke shows the fall of pressure after the exhaust valve is opened.

Diagram IV, Fig. 1, shows the piston on the fourth stroke, which is the second stroke inward and called the exhaust stroke. The exhaust valve, already opened near the end of the previous stroke, remains open throughout this stroke and in some engines does not close until a short time after the suction stroke has started. The portion of the diagram given by the indicator during this stroke is the exhaust line *x*. This line is exaggerated for the same reason as the expansion line in Diagram I. It is above the line *A-B* and the back-pressure during this stroke is from one to two lbs. according to the resistance of the exhaust passages.

It will be noted that, in the four-cycle type, there is but one explosion to every two revolutions of the engine shaft. Therefore the engine must have a large flywheel to carry it during the three strokes when there is no pressure driving the piston and to store up power sufficient to compress the mixture previous to the explosion. A modification of the four-stroke cycle was brought out in 1880 by Dugald Clerk to overcome this and secure an explosion at every revolution of the shaft. In this engine, instead of using the engine proper as an air pump (for the suction stroke) there was added a separate cylinder for this purpose. The mixture was drawn into this pump and compressed to a few pounds. The exhaust gases from the engine cylinder were discharged through an opening in the cylinder wall uncovered by the piston just before it reached the end of the expansion stroke. Just as the piston was about to return, a valve opened communication with the pump cylinder, admitting the mixture under pressure to the engine cylinder, driving out the remaining exhaust gas and filling the cylinder with a fresh mixture. This arrangement eliminated both the suction and the exhaust stroke of the four-cycle type, and completed the cycle in two strokes of the engine piston. This gave us a two-stroke cycle, and was the original of the *two-cycle engine.*

Following the introduction of the Clerk engine came a class of motors which, instead of having an auxiliary cylinder to compress the charge, employed for that purpose an enclosed crank-chamber. The crank-chamber is that portion of the motor which surrounds the crank-pin and the crank arms. It is also called the base and the crank-case. The inward stroke of the piston draws a charge of fuel and air into the crank-case through a check valve and on the outward stroke of the piston the check valve closes and the charge is compressed as in the pump of the Clerk engine. Just before the piston reaches the end of the expansion stroke, it passes an opening or port in the wall of the cylinder and in most engines of this type, a port in the opposite wall of the cylinder in communication with the crank-case is passed immediately after. In some of the earlier engines of this type instead of the port to the base a mechanically operated valve was employed. In others, many of which are in use today, instead of the check valve leading to the base, there is a third port in the wall of the cylinder which is uncovered at the end of the inward stroke by the outer end of the piston.

To distinguish the motor with the check valve from that taking the mixture into the base through a port in the cylinder wall, the first, on account of having but two ports in the cylinder, is called a *two-port* motor. That with a port admitting the mixture to the base, is called a *three-port motor*.

The pioneers of the base-compression form of two-cycle were the Nash and the Day engines. The former employed a mechanical valve to admit the gas to the cylinder, the valve communicating with the base, and the latter was the first of the three-port type. Patents were also taken out by several inventors on the three-port in the United States by Sintz, and in France and Switzerland by Soehnlein.

The operation of the two forms of two-cycle are shown in Figs. 2 and 3. The views of each motor are shown with the diagrams usually obtained from both the cylinder and the crank-case. The pressure in the crank-case is shown to a larger scale than that in the cylinder, as the pressures in the base do not generally exceed eight pounds. Similar points in these diagrams have the same letter as in Fig. 1.

Diagram I, Fig. 2, shows the two-port form of two-cycle with the piston on the inward stroke. The crank-case being air-tight, the piston draws a vacuum within it and the mixture is sucked through the check valve *V* into the base. At the same time a previous charge of mixture is being compressed in the cylinder, all openings to the cylinder being closed by the piston. On the indicator diagram, the line *s* is the suction line for the crank-case which shows a vacuum on account of being below the atmospheric line *A′ B′*. On the indicator diagram for the cylinder, the line *c* is the compression line with the ignition point *i* and the end of the compresson curve shown at *C*. It will be noted that, excepting for the absence of the suction and exhaust lines, the diagram is similar to that of the four-cycle motor.

Diagram II, Fig. 2, shows the piston nearing the end of the outward stroke with the intake port *I* from the crank-case just beginning to open and the exhaust port *X* about half open. The full arrows indicate the new mixture entering from the base and the dotted arrows, the burned gases escaping from the exhaust port. As the mixture enters the cylinder it strikes the deflecting plate *D* and is thrown toward the head of the cylinder to pre-

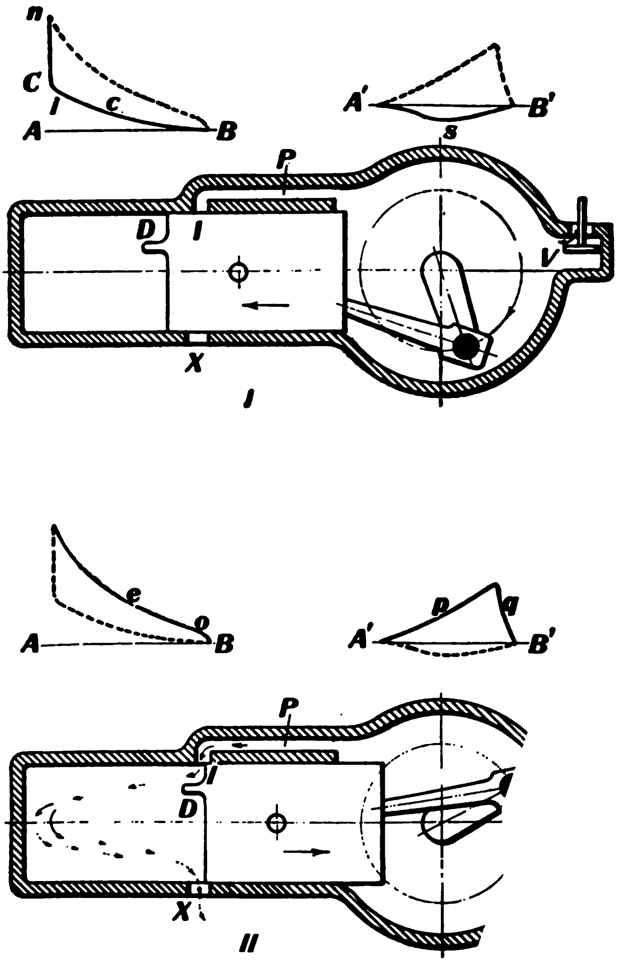

FIG. 2.

vent it passing straight across the end to the exhaust port before the latter is closed by the returning piston. The piston on its outward stroke compresses the gas and air in the base and it is this pressure that drives it through the bypass *P* into the cylinder. It should be noted that the check valve *V* is closed during this stroke.

The indicator diagram shows the compression line *p* on the diagram of the base and the sudden drop of pressure indicated by the line *q* as the piston uncovers the intake port. The diagram for the cylinder shows the expansion line *e* with the sudden change in the curve at *o*, which indicates the point at which the exhaust port is uncovered. This corresponds to the point in the four-cycle expansion curve indicating the opening of the exhaust valve.

Diagram I, Fig. 3, shows the three-port form of the two-cycle type, with the piston on the inward stroke. In this type there is no check valve and a vacuum is formed in the base, which increases as the piston approaches the end of the stroke, and until it uncovers the crank-case inlet port *K*. As shown in the diagram the piston is just uncovering this port and the fresh mixture is rushing in. At the same time the charge in the cylinder is being compressed and ignited. The return of the piston closes the port *K* so that the charge is trapped in the base.

The indicator diagrams obtained during this stroke show a drop below the atmosphere considerably greater than with the two-port form, and a sudden rise approaching closely but not quite to the pressure of the atmosphere at the end of the stroke. The compression curve is of the same form as in Fig. 2.

Diagram II, Fig. 3, shows the piston on the outward stroke with the exhaust gases escaping and the fresh mixture entering from the base as in the two-port motor. The indicator diagram is the same as for the two-port.

The Clerk type of engine is still manufactured, and a similar type, the "Koerting," is employed extensively for stationary purposes. Of the smaller two-cycles, such as those used for marine and automobile purposes, the greater number are of the closed base variety, about equally divided between the two and the three-port. While the three-port was the first of the closed base two-cycles to be proposed, the two-port was the first to be

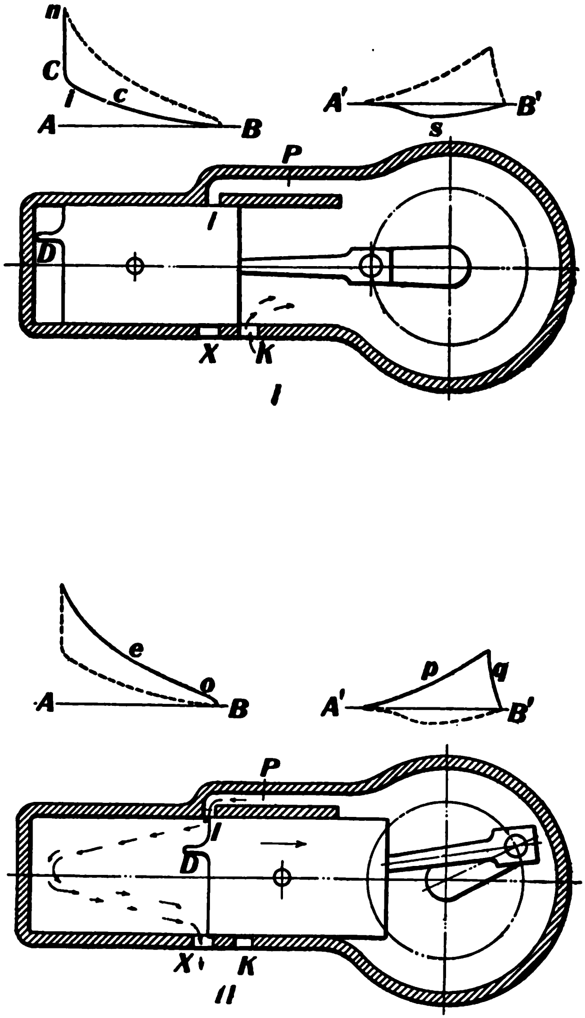

FIG. 3.

manufactured extensively. There is also being marketed what is termed a *"four-port"* motor, a combination of the two and the three-port, which is variously fitted out. In one form, the mixture is taken into the base through the check valve, and the pure air only is taken into the base through the third port. In another, the mixture is supplied through both the valve and the third port. In the first, the motor is started with the third port closed until it is up to a certain speed and the third port is then opened. The "four-port" is practically no more than a plan to increase the area of the passages carrying mixture to the base, and merely adds complications without any great advantage over the simple three-port with intake passages equal in effective area* to the sum of the two intake ports of the "four-port" motor.

There are in use quite a number of methods of supplying the gaseous mixture to the base and from the base to the cylinder as well as several forms in which the function of the air-pump is performed by enclosing the end of the piston opposite the power end. In some a rotary valve is used, either a disk on the crank with a slot registering with a slot in the wall of the base at the right time, or a tubular valve revolving in a valve cage similar to a Corliss steam engine valve. In fact, the variations of the two-cycle type are so many that it would require a fair sized volume to describe them all.

The two-cycle motor has always been attractive to the inexperienced designer or would-be inventor. It is probable that the simplicity of the motor is to him the attractive feature. He generally assumes that the motor has a number of faults which it really does not possess, and attempts to overcome what does not exist. This has resulted in many designs which are wholly the result of inference and guesswork, and have brought much adverse criticism upon the motor as a type. Properly designed and carefully constructed, the two-cycle need offer no excuses to the four-cycle.

There is still a third cycle extensively used, especially in merchant ships and submarines. This is the Diesel, the invention of Herr Rudolph Diesel, a German scientist. It was brought to the attention of the public in 1897. Theoretically, it excels the cycle of Beau de Rochas in

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*See *definition* of effective area in Chapter XXX.

efficiency and the engine itself shows a very low fuel consumption for the power developed.

Briefly described the cycle is as follows:

Pure air only is drawn into the cylinder, and this is compressed to a point at which the temperature of compression is equal to or in excess of the combustion temperature of the fuel. The compression is about 480 lbs. per square inch. Just before the piston starts on the next outward stroke, the fuel is forced in under a pressure slightly higher than that in the cylinder. The high temperature of the compressed air causes the entering fuel to take fire and burn as it enters the cylinder, just as gas burns when issuing from an ordinary gas burner. This does not increase the pressure in the cylinder and no explosion takes place. The pressure, however, does not fall so rapidly during the admission of the fuel as it rose during the compression, and the expansion curve is therefore slightly higher than the compression curve. The exhaust takes place at a pressure near that of the atmosphere, and very little power is lost through the exhaust gases escaping at a comparatively high pressure as they do in the four-cycle motor. The returning stroke drives the exhaust gases from the cylinder. This is for the four-stroke cycle Diesel. It is built in the two-stroke type as well.

A series of operations in the Diesel cycle (four-stroke) is shown in Fig. 4, which it will be noted, is in many respects similar to Fig. 1. In Diagram I, Fig. 4, is shown the piston on the first outward suction stroke, pure air being drawn in the valve *I*. Both the exhaust valve *X* and the fuel valve *f* are closed. That portion of the diagram given by the indicator for this stroke is exactly the same as in the four-cycle diagram, the suction line *s*, which is below the atmospheric line *A-B*.

Diagram II, Fig. 4, shows the piston on the suction or returning stroke, the charge of pure air being compressed to a pressure of twenty-five atmospheres or more, and in some of the earlier Diesel motors as high as 550 lbs. per square inch. Just before this stroke is completed the fuel valve is open, and the fuel, usually crude oil, is injected into the cylinder by means of air at a pressure much higher than that in the cylinder, and on account of the high temperature caused by the compres-

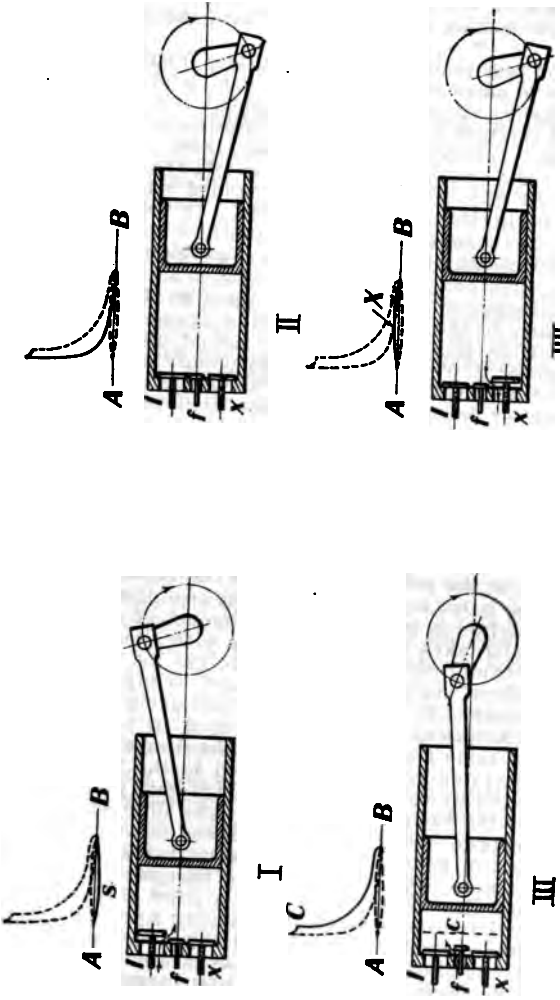

Fig. 4.

sion, it ignites on entering and burns much in the same way as gas issuing from a gas jet, until the fuel valve is closed.

At the beginning of the next stroke, Diagram III, fuel continues to enter for a short time. The heat of the burning fuel keeps the pressure up so that it falls very gradually until the fuel is shut off, when the piston has reached the point corresponding to the point C on the diagram, and during the balance of the stroke the contents of the cylinder are expanded and exhausted exactly as in the four-cycle engine. In Diagram IIII., we have the exhaust indicated by the line X, the exhaust escaping through the exhaust valve, as indicated by the arrow.

The indicator diagram given in Fig. 4 approaches more nearly to the theoretical than to the actual diagram obtained from this motor. On the actual diagram of the Diesel motor, the point C is not so plainly defined.

The Diesel engine is rapidly increasing in popularity, especially for marine work. The fact that the engine will operate on crude oil with marked success adds greatly to the low fuel cost, per horsepower hour.

In addition to the above, there is a fourth cycle, known as the "semi-Diesel." This cycle resembles the Diesel insofar as pure air only is taken in during the suction stroke and the fuel is injected at or near the end of the compression stroke. A better name is "Hot-Bulb" engine, as the use of the principle antedates the Diesel.

Ignition is not accomplished by the heat of compression, Instead the fuel is injected into the mouth of a "hot-bulb" which usually takes the form of a cup with a hemispherical end. The compression used is about 180 lbs. In the majority of these engines, no air is used to inject the fuel, which is pumped in by means of a small pump. In some of the later American designs air is used to pulverize the fuel, in the same manner as in the Diesel engine.

A new cycle recently introduced is the Brons. The series of operations and the thermodynamic principle are practically the same as the Diesel. The difference lies in the fuel supply. The fuel is introduced into a small cup containing two minute holes communicating with the compression space. The friction of the compressed air passing through these holes heats it to a point that vaporizes the lighter constituents of the fuel, causing a pressure in the cup which drives the fuel into the compression chamber. The cup eliminates the injection apparatus used in the Diesel. The cycle was invented independently by R. M. Hvid, in the United States.

CHAPTER II.

COMPARISON OF THE CYCLES.

From the designer's point of view, the four-cycle has the advantage of wider use throughout all branches of gas engineering and therefore is the better understood of the different types. It has the advantage of the longer period of development, and hence, of the greater mass of experience. So great has been the improvement in the two-cycle within the past few years that the author has been compelled completely to revise his opinions as expressed in the first edition of the Handbook. The faults of the two-cycle as enumerated in that edition have been overcome one by one, until today the two-cycle has attained a position of prominence and bids fair to surpass the older type.

In control, ease of starting, smooth running at both high speeds and low speeds and economy of fuel the present day two-cycle of first-class design is fully the equal of the four-cycle. Until recently, the four-cycle has had a great advantage over the two-cycle in fuel economy, and at small fractions of the full load it has still a small advantage. Even in this, the two-cycle is improving and some designs show a better average economy than the four-cycle. Base or crank-case explosions can be entirely avoided by means of cooling devices in the bypass, as described in a later chapter.

It is true that a great many of the old difficulties still prevail in a number of the two-cycle motors yet on the market, but the more recent designs show so great an improvement that the two-cycle has become a close competitor of the four-cycle and has the advantage of greater simplicity.

One of the greatest complaints about the two-cycle has been that two engines of this type apparently alike in every mechanical part, are entirely different in their performance. The author has found that careful construction and attention to the minutiae will eliminate this objection entirely.

So many and manifest are the advantages of more frequent impulses and less weight, both in the body of the engine and in the flywheels, that the two-cycle is rapidly gaining favor in many branches of the trade. The installation of large two-cycle stationary motors in a great steel mill, many of the units being of two-thousand horsepower each, shows the inroads it has made into the stationary field. It has some favor as an automobile power, is in widespread use for marine purposes, and has been successful as a power for airplanes. Since the appearance of the first edition of the Handbook, the author has had considerable work in the design and development of this type, which will be treated fully in later chapters.

The Diesel motor is gaining ground, where high fuel economy is of paramount importance. The engine is a grand conception theoretically, and the heat efficiency is much higher than that of any other heat engine in use. The cycle lends itself to regulation by a variable supply, and it has the peculiar advantage that the heat efficiency increases as the load is reduced. Its adaptability to use with the cheaper fuels, such as the fuel oils selling as low as two cents per gallon, give it a further advantage. We are seeing its adoption in America increase as the price of other fuels advances. As noted in Chapter I, it is rapidly gaining in favor for marine service. Quite a number of large Diesel units are now in use, and the modern submarine uses the Diesel exclusively.

The semi-Diesel is finding great favor for small powers up to 100 H.P. per cylinder. It shows a medium high fuel economy, and has the advantage of low compression pressure and low first cost. It is significant that a large number, in fact, the majority, of this type are two-cycle base compression engines.

CHAPTER III.

GAS-ENGINE FUELS.

Fuel of any sort may be used for driving a gas engine. Some, such as the solid fuels, must be transformed into gas before they are available for the gas engine, but many of the liquid and all of the gaseous fuels may be consumed directly in the cylinder without the necessity of an intermediate process. There is no fuel not in a gaseous state already that may not be transformed into gas. The garbage gathered from the streets, the waste fat of the slaughter house, the fat abstracted from wool, all of the animal, vegetable and mineral oils, have, at one time or another, been transformed into gas which could be used to drive a gas engine. Fuels such as blast furnace gas, which are difficult to keep alight under a boiler, are readily ignited under compression in a gas engine.

Of the liquid fuels available in the average motor without previous treatment, there are gasoline, benzine, kerosene, distillates and alcohol. All of these can be employed in an engine fitted for gasoline, but to secure the highest efficiency, certain changes in the motor are necessary to adapt it to each fuel. For example, the compression of a motor using kerosene must be lower than for gasoline, and water should be taken into the mixture, while for alcohol, the compression should be higher.

The amount of power to be derived from a certain quantity of fuel is always greater by a large amount when the fuel is used to drive a gas engine than when the fuel is consumed under the fire box of a steam boiler and the power derived from a steam engine. Even with coal as fuel, if the coal be made into gas and used to drive a gas engine, it is found that where the very best steam engines develop one horsepower for one hour on 1½ lbs. of coal, the same amount of coal has given 1.8 horsepower when transformed into gas and used in a gas engine. Hence a gas engine will operate on as little as 0.8 lb. of coal per H. P. hour.

The gas producer which is required for the use of coal in connection with the gas engine is much simpler than the steam boiler and requires less attention. Fuels which are difficult to burn under a steam boiler may be turned into gas in a gas producer.

The value of a fuel for use in a gas engine is determined principally by the number of heat units produced when a quantity of fuel is burned. There are several other considerations such as the explosion pressure that may be derived from the fuel and the percentage of the whole that may be consumed in engine cylinder. In the process of making the gas, a percentage of the heat is employed in the liberation of the gas and therefore cannot be transformed into power in the engine.

The heat values of the various fuels are given in the table on the next page. It will be seen from the table that the greatest heat value of the available gases is that of natural gas. This is due to the large proportion of marsh gas. The heat value of natural gas varies even when taken from wells close to one another. It will average about 1,000 B. T. U. per cu. ft., although it has gone even below 600. The artificial gases such as are used for lighting average about 600 B. T. U. per cu. ft. The fuel oils and their derivatives, gasoline, kerosene, etc. average about 20,000 B. T. U. per lb., so that the heavier fuels have the greater heat value per gallon.

The heat value per cu. ft. is no measure of the value of the fuel as a power producer when considered from the standpoint of the mean pressure to be derived in the engine cylinder. Gases of low heat value do not require so much air for complete combustion and therefore more gas can be drawn into the cylinder. For example, the proportion of gas to air with producer gas, is about 1 to 1. With natural gas, the proportion is about 1 of gas and 8 of air. Therefore, one volume of natural gas is taken into the cylinder to 4½ volumes of producer gas. With producer gas at 150, there would be 675 B. T. U drawn into the cylinder to 1,000 B. T. U. of natural gas. These proportions are selected for illustration of the problem, and the actual ratios will vary with the fuels.

Of the fuels the petroleum products give the most power in the same size engine. For example, under the same conditions about 10 per cent more is derived from

Heat Values of Fuels.

Fuel	B. T. U. per lb.	B. T. U. per cu. ft
Hydrogen at 32° F	62,100	347
Carbon	14,500	
Carbon Monoxide	10,150	338
Penn. heavy crude oil	19,210	
Caucasian heavy crude oil	19,440 to 21,168	
Petroleum refuse	19,700	
Anthracite air and water gas		386
Bituminous air and water gas		324
28-candle-power illuminating gas		950
19-candle-power illuminating gas		800
15-candle-power illuminating gas		620
New York City water gas*		710.5 Ave.
London coal gas		668
Gasoline and its vapor	21,900	690
Benzine C_6H_6	18,448	
Ethylene C_2H_4	21,330	1,666
Marsh gas (methane) CH_4	23,517	1,050
Natural gas, Leechburg, Pa		1,073
Natural gas, Pittsburg, Pa		917
Acetylene C_2H_2		1,685
Semi-water gas		185
Producer gas		150
Dowson gas, anthracite		144
Dowson gas, bituminous		108
Oil gas, Pintsch		1,320
Blast furnace gas		100

---

* Carburetted water gas at 60° F. and at 30″ water pressure.

the same engine using gasoline than when it is using natural gas. However, natural gas will stand a greater compression than is practical for gasoline, and the difference is somewhat overcome in the high compression motor.

The choice of a fuel depends to a great extent on the first cost of the fuel itself. The expense of preparation is an item to be considered, such as the outlay for gas producer apparatus, etc. Another consideration is the matter of convenience in the use of a fuel. Gasoline although slightly more costly than kerosene in all localities is more convenient to use. In some districts, no gasoline is available or it may be double the cost of kerosene. On the Pacific Coast the distillates are used to a great extent as they are low in price and easily available.

For the solid fuels, the first cost of preparation for their use, their availability and cost of the fuel itself, must all be taken into consideration.

CHAPTER IV.

MECHANISM OF THE GAS ENGINE.

The majority of gas engines in use are of the single-acting type. That is, they receive the power impulse at one end of the piston only. For large powers, quite a number of stationary engines have been built of the double-acting type, receiving the power impulse on both ends of the piston as in the usual form of the steam engine. As it would be beyond the scope of this work to illustrate and describe every variety of mechanism employed on gas engines of the various types, the author will show a horizontal section through a conventional four-cycle of the horizontal stationary type in order to illustrate the relations of the various parts. All the principal working parts are brought into one plane so that they may be shown on one view. The proportions of the parts and their customary relations with each are distorted to a certain degree for the purpose of clearness.

The principal parts shown in Fig. 5 are the cylinder *C* which is surrounded with a space *W* containing water and known as the water-jacket. The piston *P*, acting alternately to draw the gas into the cylinder and to receive the power impulse from the explosion. The connecting rod *w* through which the power impulse is transferred to the crank-shaft *S*, and the engine bed or frame *A*.

The crank-shaft revolves in the main bearings *B*. On the end of the shaft outside the main bearing is the flywheel *F* to store the power between impulses. Outside the flywheel is the pulley *p*, from which the power is taken by the belt. The inlet valve is shown at *I* and the exhaust valve at *X*. As these valves must be opened but once during two revolutions of the crank-shaft, the valve operating mechanism must be driven at just half the speed of the crank-shaft. The valves are opened by the cams *L* and *K*, these cams being fastened to the cam-shaft *N*. The cam-shaft gears *G* and *g* drive the cam-shaft at half crank-shaft speed, the gear G having twice the number of

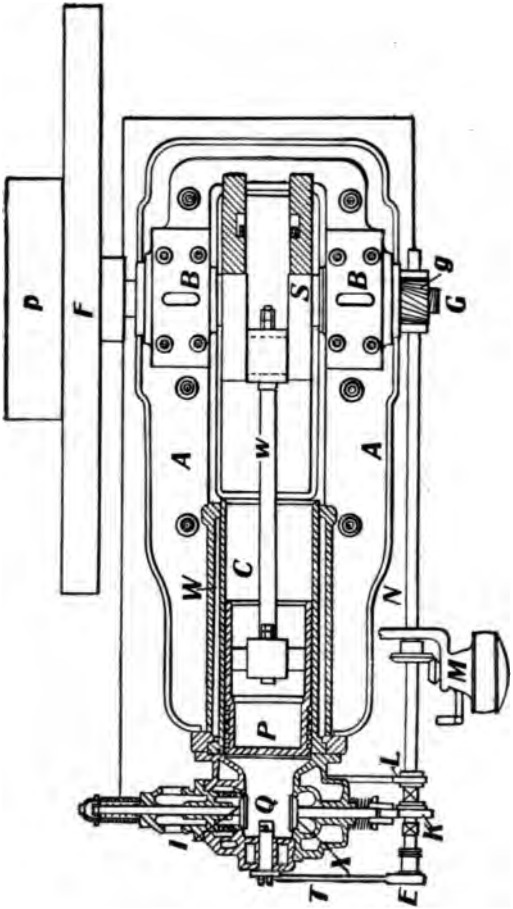

FIG. 5.

teeth as the gear *g*. The ignition mechanism must also operate but once during two revolutions of the crank-shaft. It is therefore driven from the cam-shaft. The igniter is shown at *Q* and will be described, in detail, later. It is operated by the trip rod *T* which receives its movement from the eccentric *E*. The governor *M*, which usually is vertical, is turned one quarter of a circle to bring it in the plane of the section. It is usually driven by gears from the cam-shaft.

In another chapter the author will show a section of a vertical motor of the four-cycle type, illustrating another typical arrangement of mechanism. The various parts of the mechanism will be shown in detail in chapters devoted to the various parts.

The mechanism of a conventional form of two-cycle is shown in Fig. 6. The engine shown is the marine type, this type being selected owing to the broader use of the two-cycle in the marine field than in any other. No attempt is made to show any particular make, but simply to illustrate the general construction and the relation of the parts.

The relation of the piston, connecting-rod, crank-shaft and cylinder are the same as in the four-cycle motor. The functions of the port, the bypass and the valve of the two port type has already been discussed in connection with Figs. 2 and 3, Chapter I. The exhaust manifold *X* contains the passage for the gases after they leave the exhaust port *x*. The intake manifold *I* carries the gas from the gas valve or carbureter to the crank-case port *i*. The pump *C* supplies the cooling water to the jacket *W*. The gears *Gg* drive the timer *T*. As the timer controls the *time* at which the spark must occur in the cylinder and since in the two-cycle motor, a charge of gas must be fired at each revolution of the crank-shaft, therefore, the timer revolves at the same speed as the crank-shaft and the gears must have the same number of teeth.

In dotted lines there is shown a typical check valve such as is used for a two-port motor. In case both this valve and its carbureter are used in the same motor with passages leading to the port in the cylinder wall, we have one form of the so-called four-port motor. The principal parts are signified by the same letters in Fig. 6 as in Fig. 5.

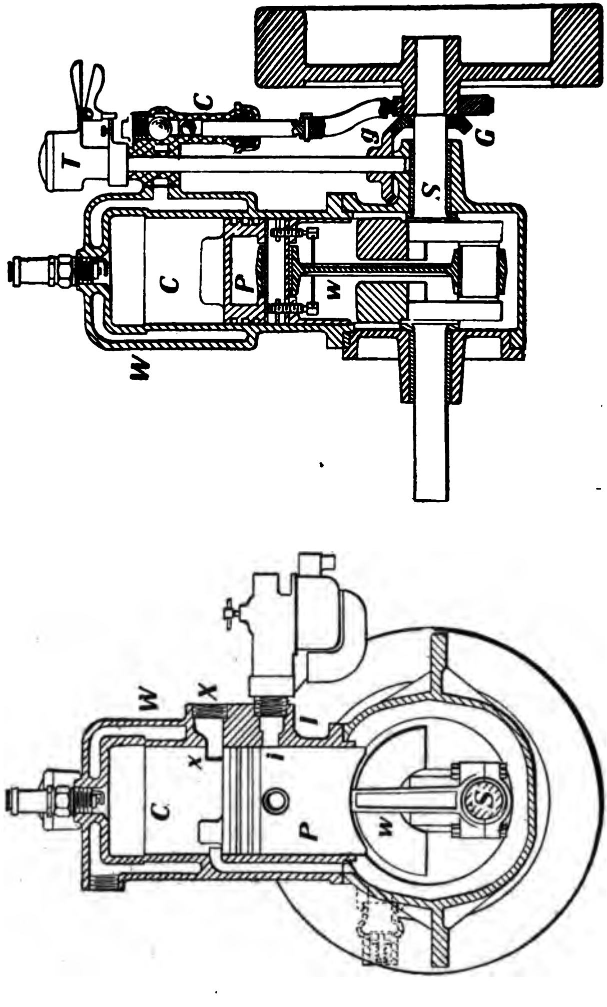

FIG. 6.

While in minor details, the mechanism of the engine may be altered for change of fuels, the general plan is the same, no matter what the fuel employed. Valve mechanisms, for example, that are suited for gas, are also suited for gasoline, distillate, alcohol or kerosene.

CHAPTER V.

VALVES, TYPES AND MECHANISM.

The valves to be discussed in this chapter, are the intake and exhaust valves, and the mechanism employed to operate them. The customary form of valve and the only form in general use is the mushroom type. It is made in two shapes as shown in Fig. 7. The conical seat *A* is that most employed, while the flat seated valve *B* has many adherents, especially for automobile motors. While it has been the author's custom to use the conical seat valve in his designs he can see no mechanical advantage in either shape. Both seat well, are easy to make, and are equally affected by pitting and dirt. Of the two, the flat valve gives the greater area of cylindrical opening for the same lift, but the direction of the gases is given a quicker change with shape B than with shape A.

The reason that the mushroom form is most used is because there are no rubbing surfaces and therefore very little wear. The high temperature within the cylinder of the gas engine makes it extremely difficult to keep the valves of either the flat, sliding, or rotary type to their seats with sufficient pressure to wear well. Many forms other than the mushroom type have been tried, only to be abandoned. The only one of this form to survive the first experiments to the extent of commercial use is the sleeve form in which a sliding sleeve surrounds the piston, opening and closing ports in the wall of the cylinder, in the same way that the piston of the two-cycle motor opens and closes the ports in its cylinder wall.

The opening and closing of the valves is now almost invariably performed by cams on a cam-shaft, rotating at one-half the speed of the engine on the four-cycle type, and where used on the two-cycle, at the same speed as the engine. The ingenious mechanisms of former days used as substitutes for the cam-shaft and gears are seldom seen today.

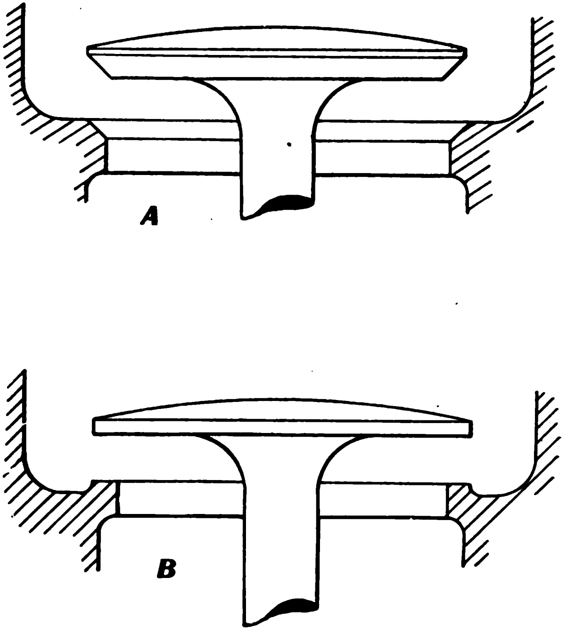

FIG. 7.

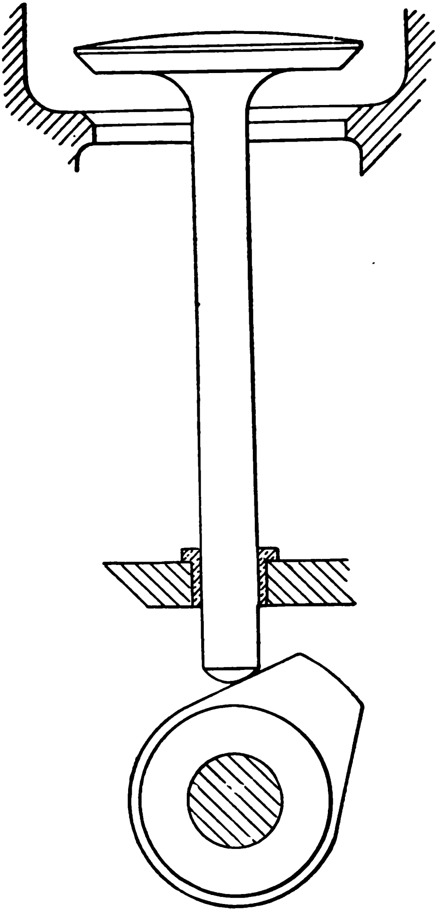

FIG. 8.

The valve is forced from its seat by the action of the cam and returned to its closed position by the action of a spring. In some engines no cam is used to open the inlet valve, but it is opened by the suction of the piston during the suction stroke. In other engines, both the exhaust and the inlet valves are opened by cams.

When the inlet valve is opened by the suction, it is called an *automatic* valve, and when the inlet is opened by a cam it is said to be *mechanically operated.* If the inlet valve is automatic, a very light spring must be employed to avoid wiredrawing (choking). It has the merit of cheapness, but the action of the automatic valve is uncertain, it sticks easily, and in a fast running engine it must be so light that it is weak. It costs but very little more to operate the inlet mechanically and modern practice is going almost wholly this way, especially on the larger sizes. Obviously the most simple method of imparting motion to the valve from the cam would be to allow the end of the valve stem to rest directly on the surface of the cam as illustrated in Fig. 8. If there were no adjustment required either for the convenience of the assembler or for wear, and if the end of the valve were broad enough to stand the wear, this would make an ideal arrangement. In fact, the author knew of a very good engine which had just such a valve mechanism.

A further step in the development of valve mechanism is shown in Fig. 9. In this, the block *B* is attached to the end of the valve-stem *S* and locked by means of the nut *N*. On the end of the block is the roller *r* which bears against the cam *C*. The objections to the plan shown in Fig. 8 are overcome as the roller takes the wear and the thread permits adjustment. The block *B* is either square or has a groove with a key or feather, or a pin to keep it from turning so the roller will rest properly on the cam. In case the center *c* of the cam-shaft is not in line with the center line *xy* of the valve stem, it should be offset in a direction opposite to the approach of the cam as indicated by the line *wv*. Recent practice is to make the line *wv* coincide with *xy*. If the center of the shaft is offset in the opposite direction and on the same side as the approach of the cam, it will cramp. The objection to this form is that it is not easy to assemble.

If the block *B* of Fig. 9 be separated from the valve

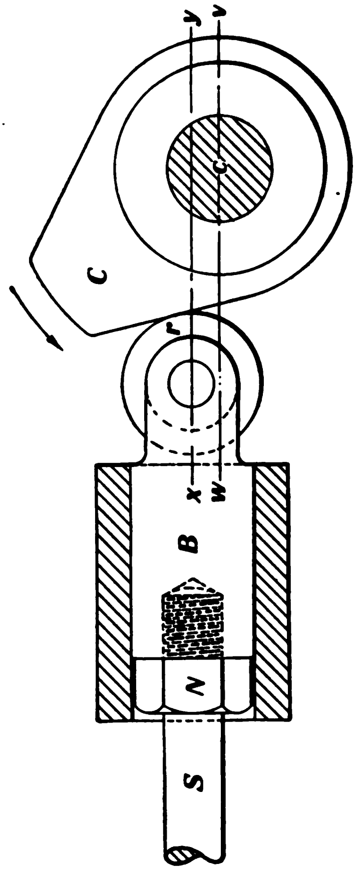

Fig. 9.

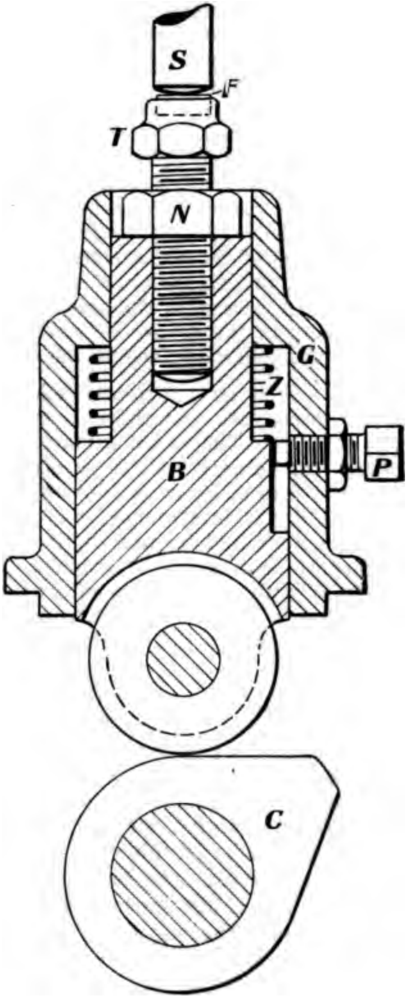

FIG. 10.

stem it is called a push rod. In Fig. 10 is shown one of the latest designs of push rod, taken from recent automobile practice. The push rod *B* slides in the guide *G* and is held against the cam *C* by means of the spring *Z*. On the top of *B* is a tappet *T* which is adjusted by screwing it up or down in the push rod, and on the top of *T* is a block of fiber *F* to take the blow of the valve stem. The fiber together with the spring *Z* give a very silent movement as the roller never leaves the face of the cam and the customary click of the tappet against the valve stem is overcome by the fiber. The space between the end of the tappet and the end of the valve stem is just sufficient to clear and allow for the expansion of the stem from the heat of the engine. A groove in the side of the push rod and a screw *P* fitting in the groove prevent the rod from turning, and keep the roller in place on the cam.

Fig. 11 shows a very good form of push rod in which the roller is dispensed with and there is no necessity of the guide pin *P*, Fig. 9. The broad surface of the mushroom head wears exceedingly well if carefully hardened, and it turns a little way at every revolution of the cam so that it wears evenly. It is stronger than a roller push rod of equal proportions and its use eliminates the roller pin, which is a continuous source of trouble. The author has used this form with excellent results on quite large engines. It requires a cam with a narrower point than the roller as will be shown in the chapter on valve design.

Where the cam is in such a position that the movement cannot be transmitted directly to the valve, levers are resorted to. These are of various forms and combinations of two or more levers are employed, according to the special function of the valve and its control from the governor. For example, in the smaller types of marine or automobile motors, the designer may choose to place the valves in the cylinder head. In this case, as illustrated in Fig. 12, it is necessary to use some form of lever arrangement, of which two forms are shown. That shown in I has the fulcrum between the valve and the rod which is pushed upward by the cam *C* when it is desired to open the valve *V*. This form requires a rod of considerable stiffness as it is in compression when opening the valve. When the rod must be long, it is a good plan to place it

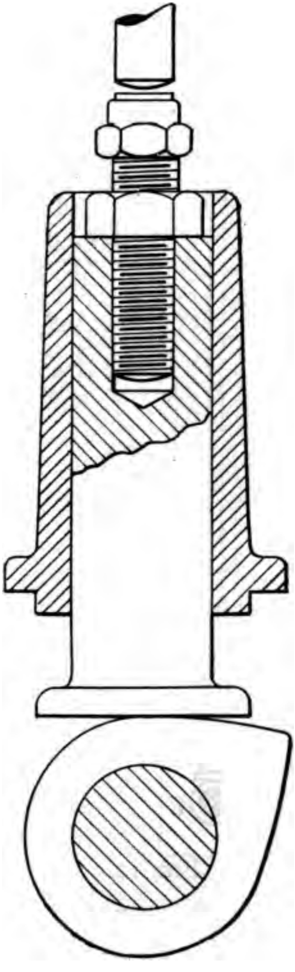

FIG. 11.

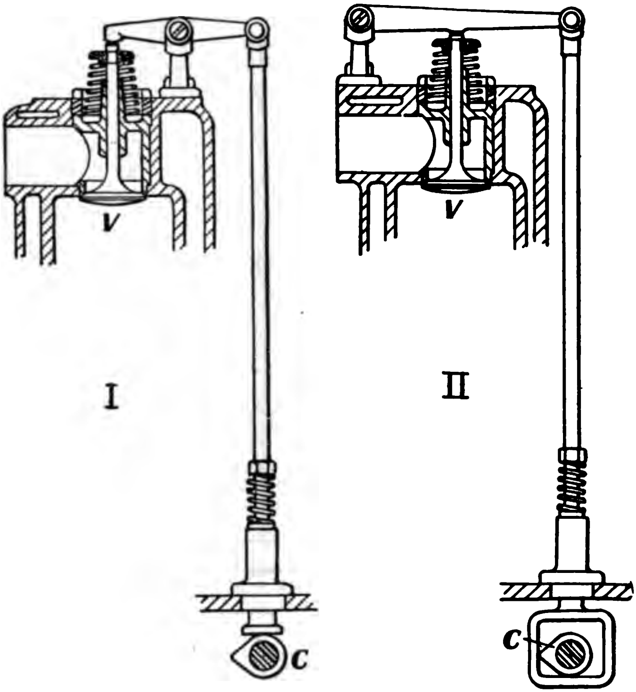

FIG. 12.

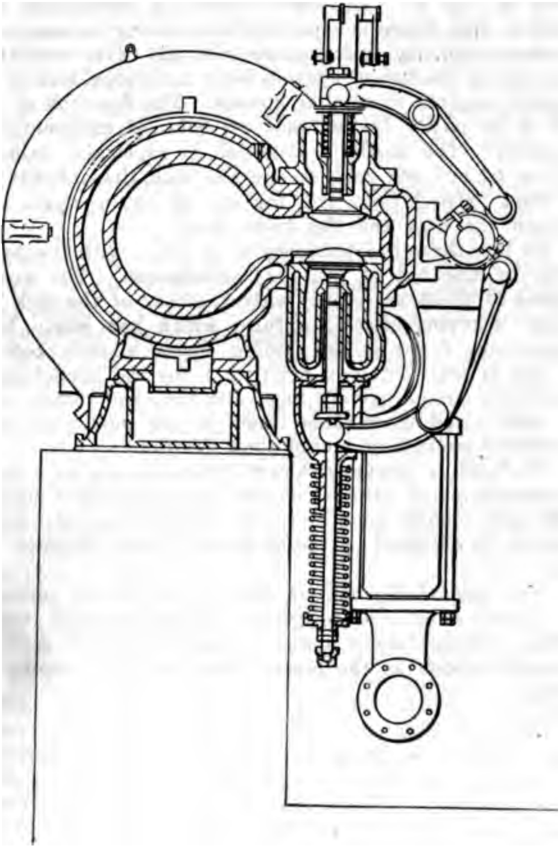

FIG. 13.

in tension as shown in II, with a stirrup having the cam inside.

The use of the lever in larger stationary engines is shown in Fig. 13. The valve operating mechanism illustrated in this figure is especially interesting because of its extreme simplicity as compared with the valve mechanism employed on the larger engines built in Europe, and in fact on many engines of smaller powers. The function of each part is so easily traced that no detailed explanation is necessary. The reader's attention is especially called to the use of but one cam to operate both the exhaust and the inlet valve levers, and the use of but a single lever between the cam and the valve stem.

Of late there is a tendency to do away with the poppet valve, by the use of various mechanisms. The greater number of these are rotary valves, either of the disk type or the revolving sleeve; a form which has lately found considerable favor is the sliding sleeve already referred to. All of these various substitutes for the poppet are in a measure comparatively new, although the rotary valve has been tried at various times in the past only to be abandoned because of troublesome features.

The sliding sleeve, although giving results as a means of admission and exhaust control, is complicated mechanically and equally good or better results may, the author believes, be obtained by the means of a well designed two-cycle.

The use of the cam is not by any means universal and quite a number of designers employ eccentrics instead. In fact, nothing about a gas engine shows such diversity of combinations as the means employed for opening the valves.

CHAPTER VI.

Gas Valves and Mixers.

In the first principles the mechanism required for supplying the gas to a gas engine is extremely simple. For example, when the engine is running at full load continuously, the gas may be turned into the intake pipe of the engine directly from the gas main through a fixed opening. An example of the service for which such an arrangement will answer is when the engine is running a dynamo for charging storage batteries. The *fixed* opening is regulated by a hand-controlled valve, but not in any way by the mechanism of the engine itself.

Such a system will not answer, however, when the engine must be operated on a varying load and the demand upon the gas supply is changing from time to time. For variable loads some form of gas valve must be employed which is controlled by the engine itself. In an engine having a hit-or-miss governor (see Chapter XII) a mechanically opened valve, opened by a cam for each power stroke, gives good satisfaction. This valve is so arranged that the governor prevents it from opening each time the speed rises above the normal and a power stroke is missed.

A form quite commonly used, and which can be employed with success on any form of engine is that shown in Fig. 14. This is simply a double seated valve with disk *D*, and a valve *R*, which closes the opening to the gas supply. For high speed engines, the valve *R* may be of rubber to deaden the noise. The disk need not close the passage entirely. The air passing to the engine raises the disk *D* and carries with it the valve stem *S* and the valve *R*. For very fast running engines it may be necessary to put a light spring on top of *D* to insure better control of the gas. The stem of this valve should be guided at both ends and if very long, just below the disk *D*.

Another form of gas valve which the author designed some years ago to overcome a bad case of noise, gives

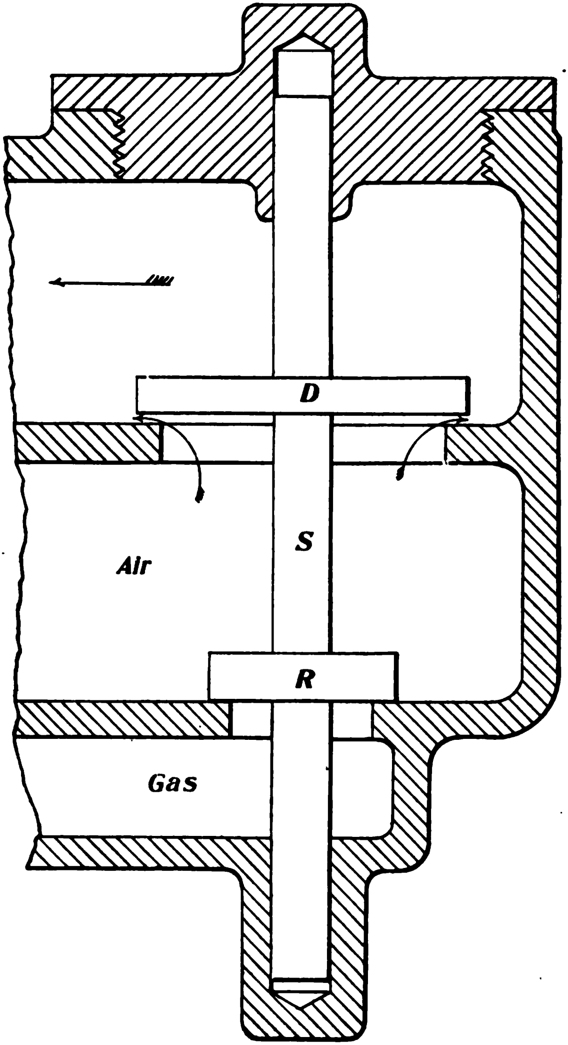

FIG. 14.

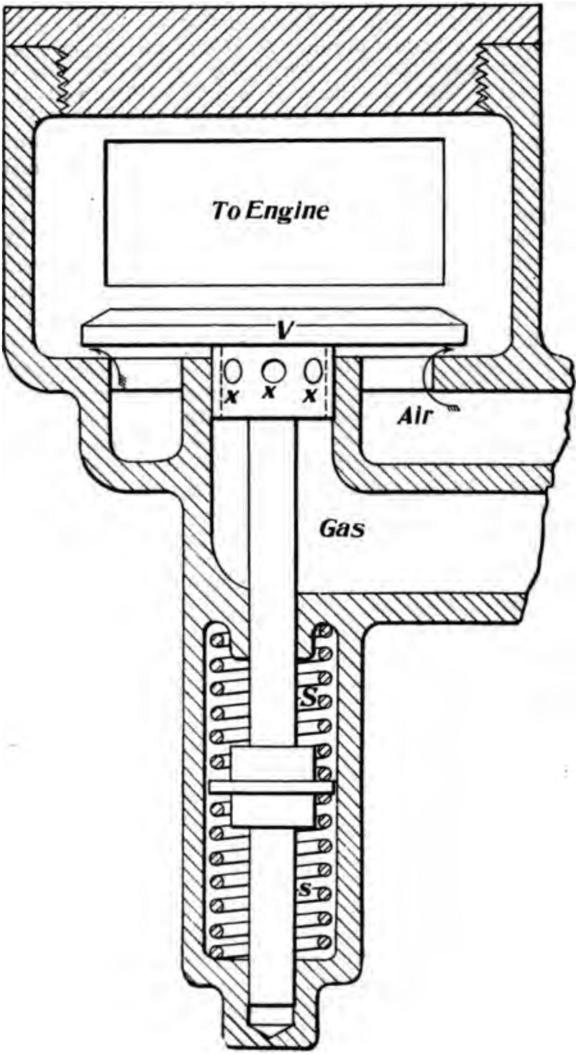

FIG. 15.

very good results and is shown in Fig. 15. The center of the valve is a hollow plunger or piston with holes *x* which come just below the top edge of the barrel when the valve is at rest. The valve head *V* is held off the seat by the spring *s* and its upward movement is restrained by the spring *S*. Air enters the inlet passage around the outside of the valve and the gas through the center. The vacuum caused by the suction stroke raises the valve, admitting both gas and air. The plunger should be free, but at the same time, it may act as a guide for the valve. What little leak of gas there may be by the plunger, between suctions, will not be of sufficient amount to disturb the mixture materially.

These two figures, 14 and 15, show the general form of a gas valve as used independently of the valve mechanism properly so called. The gas mixing arrangement is usually so intimately associated with the governing device as employed on the larger sizes of engines, that it can be taken up to better advantage when discussing the governor. In the main, the mixing device appears to invariably consist of either a double-seated valve or two valves working in unison. In each case, one valve seat opens to the atmosphere and the other to the gas supply. In a few instances, the gas and the air are controlled by one valve.

CHAPTER VII.

CARBURETERS AND GASOLINE MIXERS.

Unlike the gas mixer, the devices for mixing gasoline and air for the explosive charge of the gasoline engine often run into complications of mechanism that are more unique than useful. The most simple method of charging the air with gasoline vapor sufficient to form an explosive mixture is to draw a portion of the air across the surface of a body of gasoline or even through it. This invariably charges the air with too large a proportion of the fuel to make an explosive mixture and the charged air has to be diluted with air taken directly from the atmosphere. This method was the one first employed for the gasoline engine, but it has several objectionable features that have caused it to be abandoned.

This earlier method was known as *carburetion* and the tank or other device containing the gasoline and through which the air was allowed to pass was known as a *carbureter*. While the atomizing type in which the gasoline is drawn by suction into the air passage, was at first called a *vaporizer* to distinguish it from the tank device, "carbureter" was the term finally chosen to describe all forms of gasoline mixing devices with the exception of the mixing valve.

The *mixing valve* is of two forms, the earlier having a double seated valve, one seat a large one closing the air passage, and the other a pointed stem controlling an opening of small diameter leading to the gasoline supply pipe. The later form consists of a simple check valve with a pin-hole through the seat and communicating with the gasoline supply.

These two forms of mixing valve are shown in Fig. 16. The earlier type *I* has the air valve *V* carrying the gasoline valve *v* made from the same piece so that as the air enters from the passage about *v* and raises the valve *V* it also raises the point of the valve forming the gasoline valve *v*. The gasoline feed was regulated by the needle

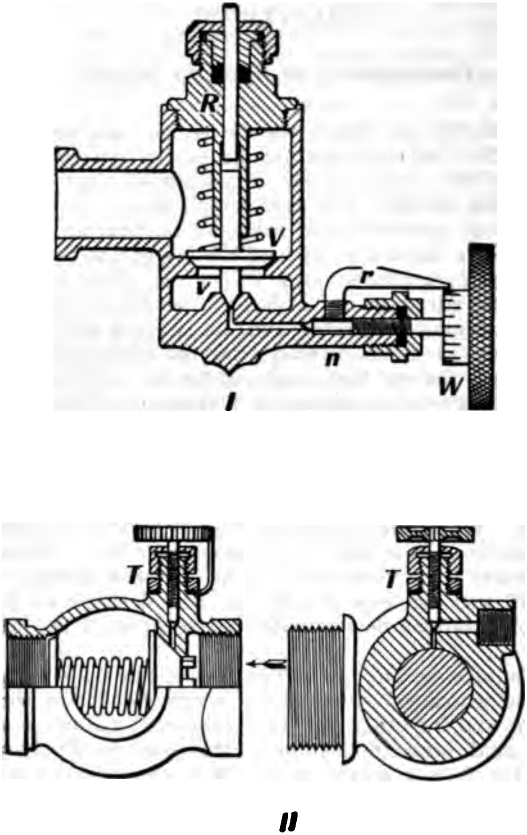

FIG. 16.

valve *n* fastened to the graduated hand wheel *W*, the pointer *r* showing how far the valve was open. The regulating stem *R* was arranged to be screwed up and down to limit the lift of the valve *V* and act as a throttle.

A later and more common form of mixing valve is the type II. This is nothing more nor less than a spring check valve into the side of the seat of which has been drilled a small hole connecting with the gasoline supply through the needle valve *T*. The air enters in the direction of the arrow and the suction of the engine opens the check valve allowing the gasoline to flow from the opening. The draft of air produces a slight suction on the gasoline which spreads over the conical surface of the valve and thus becomes well mixed with the air. This type has been used a great deal on two-cycle launch engines of the slow speed type, but is now being superseded to a great extent by one of the later forms of carbureter. A quite important point in the design and construction of this form of vaporizer is to get the needle valve as close to the opening as practical. This form of gasoline mixing device, while very simple, has several draw-backs. The needle valve has to be closed whenever the engine is stopped as the check valve cannot be made gasoline tight. Another draw-back to this device is the fact that the needle valve has to be regulated for every variation in speed of the motor.

The gasoline jet is a form of gasoline mixing device that was in considerable favor in the early days of the gasoline stationary motor, and is illustrated in Fig. 17, (page 46.) It consists in pumping a measured quantity of gasoline into the air passage and against the side of the suction pipe. It gives a fairly good success where the load is constant, or where the engine uses the hit-or-miss type of speed regulation. It has also been revived of late and is in considerable favor with some builders of aeronautical motors.

In order to understand the basic principles of the modern carbureter, the following facts must be kept in view. When air is drawn through a passage, the pressure within that passage becomes lower than that of the surrounding atmosphere. The smaller the passage, and the greater the speed of the air current, the greater will be this difference of pressure, or as it is generally known, the degree of vacuum. Suppose that into the side of this passage there is a tube leading to a reservoir containing a liquid such as gasoline. The pressure of the atmosphere being higher than the pressure in the air passage, the liquid will be forced into the passage, and be caught up by the air, mixing with it to greater or less degree. It is easy to see that if the amount of air drawn through the passage be increased, or the entrance to the passage be choked in any manner, the vacuum will be increased, and the liquid will flow more rapidly into the passage.

Presuming, now, that the passage is the intake pipe of a gasoline motor and a tube in the side of the passage communicates with a reservoir of gasoline kept at a constant level by means of a float, for example, we have the first principle of the carbureter. If the engine speeds up, the air is drawn more rapidly through the passage, and the vacuum is increased. Air, being elastic, on entering the inlet passages expands more and more as the vacuum increases. The liquid gasoline, being non-elastic, does not expand. Hence, at the higher speeds of the motor, the proportion of gasoline to the air increases. Therefore, if the adjustment of the proportion of gasoline to air is correct for low speed, the mixture will be too rich in gasoline at high speed, and some means of overcoming this tendency must be provided.

Any device to overcome the tendency of a carbureter to furnish too rich a mixture at high speed, is known as a *compensating* device. It is in devices of this kind that we find the greatest difference in carbureters. Space will not permit of describing all of them, but I will describe examples of those most generally used.

As has already been noted, if the entrance to the air passage be choked, the vacuum within the passage will be increased. Suppose that at low speed the entrance to the passage is choked by a valve of some description and as the engine speeds up, the valve at the entrance of the passage be gradually opened, either by hand or by automatic means. Then the vacuum within the passage can be kept very closely in proportion to the flow of air. In some carbureters, this valve is opened by the operator, usually being connected with the throttle, so if the throttle is opened, the choke valve of the intake passage is opened in a predetermined ratio. The automatically operated valve is held to its seat by a spring, a small initial or fixed opening is provided for low speed and as the motor speeds up, this valve is opened by the increase in vacuum within the passage. The tension of the spring is adjustable and the vacuum within the passage is kept very closely to the correct amount by adjustment of the spring. The system above described I will call the *choking compensator.*

Another system of compensation is: instead of the entire volume of the entering air passing by the nozzle (as the end of the gasoline tube within the air passage is called) uncarbureted air is allowed to pass into the intake passage in such a manner that it meets the carbureted air at a point *between* the nozzle and the cylinder. This auxiliary air opening is closed at low speed by a valve. As in the choking type, this valve may be operated either manually or automatically. If opened by hand, it is usually connected to the throttle as in the choking type. Where the valve is automatic, it may be held to its seat by a spring, or by its own weight. In several makes of carbureters of this type, the valve is a bronze ball. Generally when the valve is held to its seat by its own weight, more than one compensating valve is used, as for example, in one carbureter using bronze balls, five balls are employed. This form of carbureter I will call the *auxiliary air-port* type.

Still another form of compensation is employed, and that is by changing the adjustment of the needle valve which controls the flow of gasoline from the reservoir to a nozzle. When this system of compensation is used, alone, the needle valve is closed as the motor speeds up, and the adjustment is always manual. The control of the flow of gasoline, however, can be made automatic by using some means of obstructing the flow, as the motor speeds up. One way of doing this is to lower the level of the gasoline in the reservoir as the drain on the reservoir increases, and another is by allowing the gasoline to flow into a basin at the bottom of the carbureter so that at low speed carburetion is produced on the early principle of causing the air to pass over a considerable body of the fluid; while at high speed the basin is drained and the gasoline is compelled to flow through a small passage. This system may be called *gasoline flow control.*

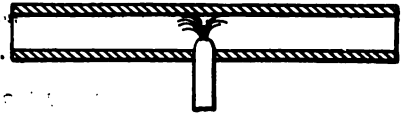

FIG. 17.

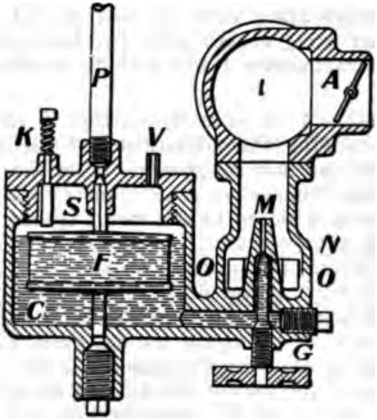

FIG. 18.

It is obvious that two or more of these systems may be used in combination. For example, in the choking type, there is a tendency with the four-cycle motor, for the carbureter to pull a richer mixture at low speed than at high speed, and to overcome this, an attachment to open the needle valve slightly as the speed of the motor increases is used. In both the choking type and the auxiliary air-port type, the adjustment for a two-cycle motor must be in the opposite direction and the needle valve opened more fully at low speed. Instead of opening the needle valve at high speed for a four-cycle motor, there is sometimes provided a series of nozzles of different heights, so as the speed of the motor increases, the gasoline is compelled to flow through the higher nozzles and more gasoline provided in this way. We also find quite a number of variations and combinations and quite frequently other means of keeping the proportion constant between the air and gasoline at all speeds of the motor. Another arrangement is to use double carbureters, a small carbureter for low speed, and a larger carbureter for high speed, either separate or in combination.

Attachments of various kinds are frequently supplied for warming the air, either before it enters the carbureter, or after it has picked up the gasoline. There is the air tube connecting the carbureter to some warm part of the engine for heating the entering air; then, again, there is the jacket surrounding the air passage, heated either by hot water from the jacket of the engine cylinder, or by transferring a portion of the exhaust gases through the jacket. Furthermore, we find dash-pots and buffer springs to overcome the fluttering of the compensating valve on its seat, and other combinations too numerous to mention.

In Fig. 18 is shown in diagrammatic form, a carbureter which illustrates the general relation of the various parts. The float chamber or constant level reservoir, *C*, is fitted with a hollow metal float, *F*, carrying a pointed stem *S* which, in its upper portion, is made either square or triangular in order to allow a passage for the gasoline between the sides of the hole in which it slides, and the stem *S*. At the upper end of this stem is a sharp pointed valve which is ground to a seat in the lid of the carbureter. Gasoline flows from the tank through the pipe *P*. A vent tube *V* allows communication with the atmosphere, some-

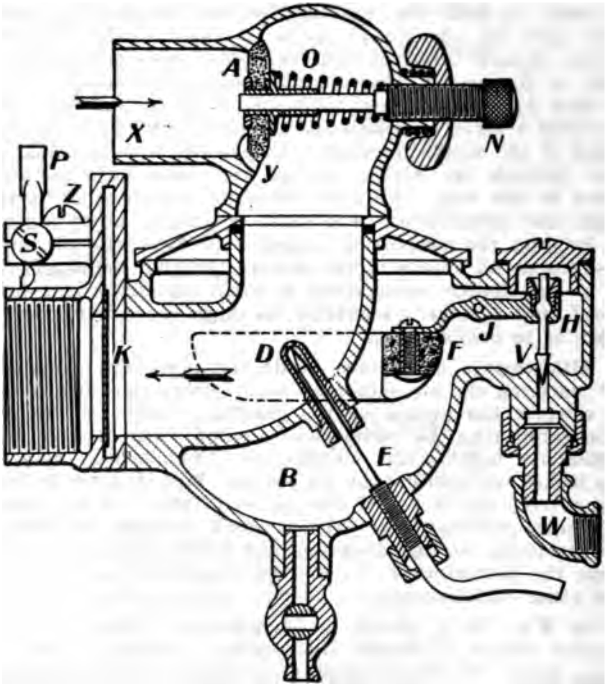

FIG. 19.

thing which is very necessary, for if the float chamber is air-tight, gasoline will be trapped within it, and will not flow to the nozzle. The opening from the float chamber to the nozzle *M* is adjusted by means of the needle valve *G*. Air entering at *O* due to the suction of the motor, passes the nozzle *M* and, as already explained, the vacuum in the passage surrounding the nozzle causes the gasoline to be forced through the opening in *M* by the pressure of the atmosphere on the body of the gasoline in *C*. In passing from the nozzle, the gasoline is usually broken up in a fine spray and mixes with the air, and then the air and gasoline vapor together are drawn into the motor through the passage *I*. At *A* is shown the hand operated valve which is opened gradually as the speed of the motor increases, as already explained.

When cranking the motor, the air does not always enter the carbureter with sufficient speed to draw the gasoline through the nozzle end, and in order to get sufficient gasoline for the initial explosion, the float *F* is pushed down by the priming pin *K*, raising the level of the gasoline in *C* until it flows over the top of the nozzle, and settles in the small depression below the openings *O*. Instead of using the priming pin *K*, some carbureters are provided with means for choking the openings *O*, thus increasing the vacuum around the nozzle at low speed.

In Fig. 19 is shown a carbureter with a choking compensator. All the air passing to the motor enters at the opening *X*. This opening is almost entirely closed by the leather valve *A* held against its seat by means of the spring *O*. There is a small initial opening *y* through which all the air must enter until the suction is sufficient to lift the valve *A* from its seat. As already explained, the increase of the suction raises the valve from its seat; then, by adjusting the pressure of the spring by means of the thumb screw *N*, the vacuum surrounding the nozzle *D* is kept at such a point that the proportion of gasoline and air is practically constant.

As this carbureter has several features peculiar to itself, it will be interesting to describe them. Gasoline enters the carbureter through the elbow *W* passing upward by the float valve *V* which is closed when the gasoline in the float chamber *B* lifts the cork float *F* to the height shown in the figure. The arm carrying the float F and the valve V is pivoted at *J*. A ball and socket joint will be noted at *H*. The needle valve *E* is held from turning by the pressure of the packing surrounding it. At *K* is the throttle, operated by the lever *P* and fitted with a stop screw *S* locked by the screw *Z*, so that the throttle is prevented from closing entirely. The point of minimum closure employed is usually that at which the engine will just barely run when the load is thrown off.

In carbureters of the auxiliary air-port type, a valve very similar to the valve *A* in Fig. 19 is employed except that in this type there is no initial air opening in the valve seat and only the compensating air passes by the valve. In Fig. 20 is shown that form of this type of carbureter, using bronze balls instead of a spring controlled valve for the auxiliary air-ports. At low speed all air enters at *G* and passes upward through the passage *Q* past the nozzle *X*. The peculiar shape of the passage at *Q* is known as a *venturi* tube and is a form which causes very little frictional resistance to the passage of the air, but, on account of the higher speed of the air at its narrowest portion, however, it forms a little more vacuum around the nozzle than would ordinarily be produced.

The compensating air enters through five passages as indicated by the dotted line whenever the suction is sufficient to lift the balls *K*. In one form of the ball-valve carbureter, the balls vary in size, the smallest and lightest ball lifting first, or when the suction is least, and as the suction increases, each heavier ball is lifted in turn. In the carbureter shown in the illustration, the balls are all the same size, but the taper of the seat is different for each ball. Therefore, each ball presents a different area to the air, and that which has the largest area presented to the air is the one that will lift first. It will be noted that while the carbureter shown in Fig. 19 has two adjustments, the tension of the air valve spring being one, and the opening of the needle valve being the other; the carbureter shown in Fig. 20 has but one adjustment, that of the needle valve *A*. It should also be noted that there is a small basin surrounding the lower end of the needle valve to assist the air in picking up gasoline at low speed. The opening *G* is usually threaded so that it can be easily connected with a hot air pipe for cold weather and heavier fuels. The float valve mechanism is the same in principle as that shown in Fig. 19. The throttle, of the butterfly type, is shown at *E*, and is operated by the lever *B*. The connection to the motor intake pipe is made at *F*.

In any form of carbureter there is usually some initial adjustment or alteration to be made to adapt it to one particular form of motor. For example, the spring *O*, in Fig. 19 must be lighter for some motors and heavier for others. In some others the size of the initial air opening *y* may be changed to advantage, and occasionally, but not often, a motor will be found that works best with no initial air opening at all. In the carbureter shown in Fig. 20, the size of the opening surrounding the needle valve must be varied either by bushing it to make it smaller or boring it out to make it larger. Sometimes the lift of the valves *K* has to be increased slightly by placing a gasket or washer underneath the caps *k*.

This initial adjustment to one type of motor is best made by a specialist from the carbureter factory. If the carbureter works well at high speed, but does not draw sufficient gasoline at low speed, it is necessary to increase the tension of the spring *O* on the type shown in Fig. 19, or to decrease the size of the opening *Q*, in Fig. 20 and of course, the opposite is true for the opposite condition. In the carbureter in Figure 20, if it draws too much gasoline at high speed, the correction may be made by raising the caps *k*.

The reader should note that in any and all forms of compensating mechanism, the automatic compensation given by the spring controlled or the weighted valve meets general carbureter conditions much more perfectly than the manually operated valve, and for the following reasons. Take, for example, an automobile motor climbing a good grade with the throttle wide open, and yet the engine turning over slowly. The speed of the entering air will be necessarily much slower than with the throttle wide open and the engine running at high speed, and yet, if the compensating valve is connected to the throttle it will be set in the position best for high speed and wide open throttle, and will not give a mixture sufficiently rich in gasoline. On the other hand, a carbureter which compensates by means of an automatic valve meets this condition exactly as well with a wide open throttle at low speed, as the motor running at high speed. It

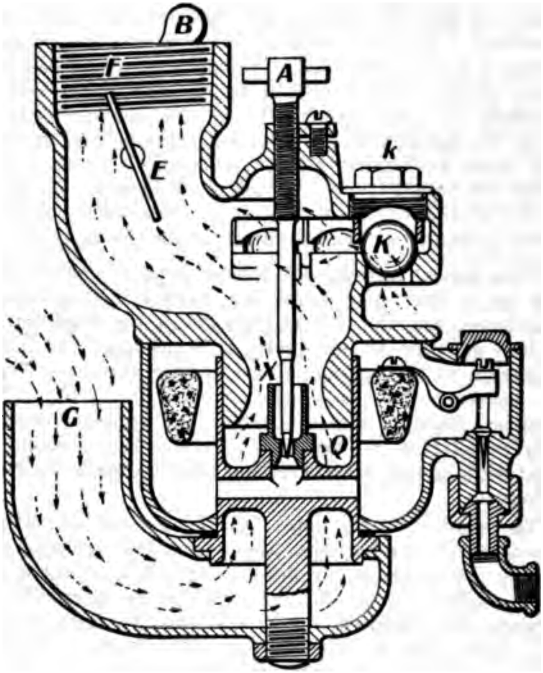

FIG. 20.

should be noted, however, that any form of automatic compensating valve produces a choking effect and the motor will not get as much air, particularly at high speed, as it would with the manually operated compensator. For this reason the hand operated valve allows the motor to give greater power at higher speed than the automatic type. A good example of this is the fact that in a carbureter of the type shown in Fig. 19, pushing the valve *A* away from its seat by means of a rod or a pencil will allow the motor to speed up on the same resistance, as, for example, in a boat. For racing, therefore, quite a number of operators remove this valve entirely, and get sufficient vacuum for starting by choking the opening *X* with the hand. It should be remembered, however, that if the valve is removed, the motor will not operate satisfactorily at low speed, except by increasing the opening of the needle valve *E*. It is the writer's experience that the type shown in Fig. 20 gives slightly less resistance to the entering air than the type shown in Fig. 19, but a great many operators claim that they do not get satisfactory results with this type. As a matter of fact, the selection of a carbureter that will give the best results on any motor depends a great deal on what service is expected of the motor. It should be remembered that in order to get the very greatest amount of power at high speed, regulation at low speed must be sacrificed, and for the best regulation at low speed, the power of the motor at high speed must be sacrified.

CHAPTER VIII.

OIL ENGINE VAPORIZERS AND MIXERS.

As we pass from gasoline to the heavy fuels which do not vaporize readily at ordinary temperatures, we find complications more or less extensive in order to secure thorough vaporization. The basic principle of these devices is to assist vaporization by heating the fuel. This heating is accomplished in various ways, ranging from drawing hot air into the carbureter, to breaking up the oil into a fixed gas, by means of the oil gas producer. Generally, the methods may be classed as follows: Preheating the air; the use of the hot bulb or a hot point in the cylinder; the use of extremely high compression to raise the temperature of the air, as in the Diesel cycle; or generating the oil into a gas by means of an oil gas producer. For the fuels intermediate between gasoline and kerosene, such as the California distillates, it is usually sufficient to start the engine with gasoline and allow it to get warm, and then change to distillate. As a general rule on motors of small bore and stroke, priming with gasoline is sufficient. The writer has found, however, that on some motors priming with distillate and using no gasoline whatever, will start even a cold engine. Warming the air entering the carbureter by drawing a part of it from around the exhaust pipe will help in the vaporization of distillate, and even kerosene.

For kerosene, it is usually the better plan to run for a short time on either gasoline or distillate until the engine is thoroughly warm, and then change to kerosene. However, I have found that priming with gasoline is often quite sufficient to start a small two-cycle motor on kerosene. The use of warm air improves the vaporization of this fuel, and on the heavier fuels, especially on kerosene, the motor should be allowed to run hotter than on gasoline, as it is generally considered that the temperature of boiling water is that best for the jackets of a kerosene motor. Another very good plan for the operation of a kerosene motor with the ordinary gasoline carbureter is to coil the fuel oil tube leading to the carbureter around the exhaust pipe. This has proven particularly effective on kerosene motors operated in Arctic waters. In starting the engine the coil is first heated by means of a torch.

In recent practice the use of water in the mixture is considered necessary when operating motors of the ordinary two or four-cycle type on kerosene. The water is drawn into the carbureter or the inlet passages in various ways, one of the most successful being by means of a small carbureter attached to the fuel carbureter, and called a *water carbureter*. The water reduces the sharpness of the explosion and overcomes the knocking experienced in the use of kerosene on motors having comparatively high compression. The quantity of water required is easily determined by the performance of the engine.

Quite a number of motors operated on kerosene, and even heavier oils are so arranged that the cylinder has a compartment of small volume, not water-jacketed, which is brought to a dull red heat and kept at this heat by the combustion of the charge, and the fuel is injected directly against the side of this compartment. This is what I have termed the *hot-bulb method*. In the *hot-point* or *hot-plate* method the piston carries a set of points or plates usually made of platinum and kept at a high temperature by the combustion within the cylinder. The fuel is injected through a spraying nozzle against this plate. The author has seen this method employed on crude oils so heavy and thick that they could scarcely be pumped. In the Diesel cycle described in Chapter I. pure air only is compressed to a high pressure, and a high temperature, and the fuel oil vaporizes and ignites as it enters through a spraying nozzle, and comes in contact with the heated air.

The oil-gas producer is operated by the heat of the exhaust, forming of it a more or less permanent gas, and from this vaporizer it is drawn into the motor and combined with the air to form an explosive mixture.

CHAPTER IX.

MAKE-AND-BREAK IGNITERS.—LOW TENSION MAGNETOS.

In modern gas engineering, electric ignition has practically superseded every other form excepting on some of the oil engines. In the first edition of the Handbook, considerable space was given to the hot tube igniter, which is now but rarely seen. This, and some of the special forms of ignition other than electrical, will be described in Chapter XI.

All electric ignition is produced by one or another form of the electric arc. The reader is doubtless familiar with the spark that invariably follows the breaking of an electric circuit as when a switch is opened, or when a wire connected to one pole of a battery is drawn across the other pole. This is the type of spark that is produced in the form of igniter known as the *make-and-break* type, meaning to make the circuit and then break it, and it is the type of ignition that is employed on a low tension circuit, that is, on a circuit in which the electric pressure, known as the voltage, is so low that it will not cause a spark to leap across a gap or opening. Note carefully the difference between this type of ignition and the jump spark type, as described in Chapter X. The author suggests that these two chapters be read more or less in conjunction so that the reader may the more readily grasp the difference between the two types.

The spark in the make-and-break system is usually obtained from a battery of dry cells, a storage battery, or a low-tension dynamo or magneto, giving a difference of electrical pressure of from about six to ten volts. This would mean in a battery of dry cells, four to eight cells in series, in a storage battery from two to four cells in series. As a general rule, a battery of six dry cells in series or three storage cells in series is employed, or with the Edison type of cell employing a solution of caustic soda from five to six cells in series, giving a pressure of approximately one volt per cell.

If the ends of two wires connecting the terminals of a battery or dynamo of this voltage be brought together and separated, quite a strong spark is produced. The volume of a spark produced from a source of current of such a low voltage is increased enormously if the current be made to pass through a coil of insulated copper wire wound around an iron core.

The added intensity of the circuit due to the presence of the coil, is caused by a reaction of the current upon itself. In order that this may be easier understood, I will explain briefly the relation between magnetism and electricity. From the north and the south pole of any magnet there is a field of influence into which, if any particle of iron or steel be brought, it will be pulled to the end of the magnet. The magnetic strength of this field grows weaker the greater the distance from the pole. In order to better understand a magnet, the scientist considers this field of influence as being traversed by a great many imaginary lines, called magnetic lines of force, and this field of magnetic influence is known as the magnetic field. This magnetic field surrounds the magnet and is strongest at the poles. If a wire, the ends of which are in some way connected, is passed across the magnetic field so that it cuts the lines of force, a current will be produced in the wire. Every wire through which an electric current is passing is surrounded by a magnetic field, no matter whether the wire be of iron, copper, or anv other metal. If a wire through which a current is passing is wound around a bar of iron, its magnetic field is accentuated, and the bar becomes a magnet which is stronger, the greater the current passing through the wire, and the greater the number of turns of the wire. A wire surrounding a magnet lies in a position that cuts the lines of force passing from pole to pole, as shown in Fig. 21, (page 59.) If instead of the wire cutting the lines of force, as previously described, it is not moved in the field, but the magnetic field itself is weakened or strengthened, a current is produced in the wire.

Having this reaction of the magnet upon the wire, and of a wire through which current is passing upon the magnet, fixed in our minds, we can see that if the current strength in the coil surrounding the magnet is changed, it changes the strength of the magnet, and at the same time, the strength of the field. This change in the strength of the magnetic field reacts upon the wire itself. The greater the *change* in a magnetic field passing through a wire, and the greater the *speed* at which the change is made, the greater will be the electrical pressure produced in the coil of wire. No greater change of the magnetic field can be produced, than by having it at its highest possible strength, and suddenly reducing it to zero. Now, it will be seen that when the current is broken in a coil surrounding a magnetic core, the strength of the magnet suddenly drops to zero. This sudden change of magnetic strength reacts upon the wire and produces a current of higher voltage than was produced by the battery or magneto from which the current flowed through the wire originally. This reaction is instantaneous and occurs during the *first moment* of the break, so, no matter how rapidly the circuit is broken, a reactive current of great intensity is produced, and causes a powerful spark at the gap.

The reader not familiar with electrical phenomena may not grasp this at the first reading, but the author believes that by studying it carefully, he will be able to understand just what takes place. It should be noted that the intensity of the spark is, as already pointed out, dependent as much on the quickness of the break, as on the strength of the current passing through the wire.

To secure the greatest possible intensity of current passing across the points of an igniter, the points must be clean and free from corrosion, all connections must be tight, and the wire through which the current passes from one terminal to the battery or dynamo and back to the other terminal must be of sufficient size to allow the current to flow freely. In order to break the circuit as quickly as possible, it is best to employ some such means as suddenly releasing a spring that is brought into tension when the igniter points are pressed together. It is for the purpose of impressing upon the reader the importance of these two functions of the make-and-break igniter, that I have explained so minutely the principles involved.

An extremely simple form of make and break is shown in Fig. 22, and while this particular mechanism is now very little used, it illustrates very clearly the principles

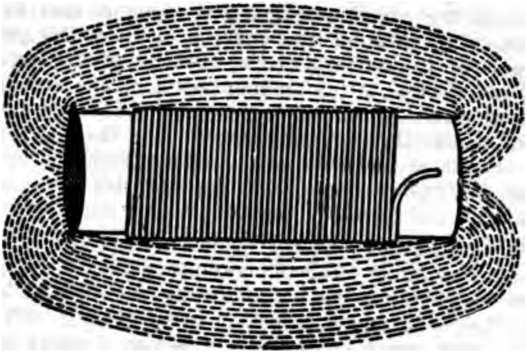

FIG. 21.

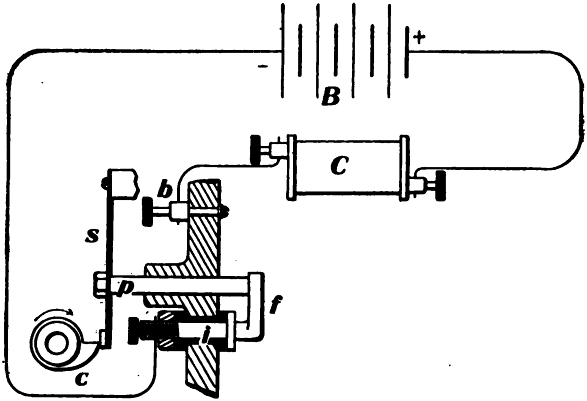

FIG. 22.

of make-and-break ignition. The battery *B* is a source of current for which a magneto or dynamo may be substituted. The path of the current is from the positive (+) pole of the battery to the one end of the induction coil *C* and from *C* to the binding post *b* on the metal work of the engine. From the metal work of the engine, the current passes through the plunger *p* to the projection *f* of the insulated electrode *i*, and thence from the binding post on the outer end of *i* to the negative (—) end of the battery.

The projection *f* of the insulated electrode is within the engine cylinder. The plunger *p* is attached at the end outside the cylinder to a flat spring *s*, which is so set that it has a tendency to pull *p* away from *f*. The end of this spring rests against the step-cam *c*, which travels in the direction of the arrow. When the projection first comes in contact with the end of the spring, it pushes *p* into the cylinder until it meets *f*. Further movement of the cam simply increases the pressure between *p* and *f* by bending the end of the spring. When the point of the cam slips off the end of the spring, the spring is released suddenly and, snapping back, pulls *p* sharply away from *f*. The consequence is a quick break between *p* and *f* and a spark of considerable intensity.

The coil *C* consists of a core of soft iron wires around which is wound several hundred turns of cotton covered wire, generally about sixteen gauge. The length of the coil varies from five to ten inches.

The length of the coil is of importance as it involves a feature in make-and-break ignition which should be carefully considered. The longer the coil and its iron core, the slower will the core reach its full magnetic strength when the current is turned into the coil around it. In a short coil the magnet is brought more quickly to a saturated state than in a long one, therefore the current need not be turned into the coil so far ahead of the time of breaking the circuit at the igniter when using a short coil as when using a long one. Consequently, for a high speed engine, a short coil should be used, generally not over five inches long. On a large, slow moving engine, a coil as long as ten inches will give good satisfaction. By low speed engine in this instance, is meant an engine *running four* hundred revolutions or less.

As all coils require a definite though minute time for the current to bring the core to its full magnetic strength, and as the intensity of the spark depends on the strength to which the magnet is brought, as well as the quickness with which the break at the igniter points is made, a certain definite time must elapse between the closing of the circuit and the time at which it is broken—in other words, between the make and the break. The securing of good results from the make and break igniter depends on a thorough understanding of the principles involved, and the making of an igniter based on these principles, and fulfilling the requirements as outlined.

For the basic principles of the magneto or dynamo, the reader is referred to books on electrical machinery, as the subject is too broad to be treated here. It may be said, however, that there is quite a little misunderstanding as to what a magneto really is. The magneto is nothing more than a dynamo in which the magnetic field is comprised of permanent magnets, while, in the dynamo, electro-magnets are used which are excited by diverting part of the current generated by the dynamo itself through coils surrounding the magnetic cores. The voltage of a low tension magneto or dynamo, for both are used for gas engine ignition, is about eight volts. The low tension magneto as distinguished from the high tension type, which will be treated in Chapter X, is one in which the current is taken directly from the armature to the ignition device, being led either directly to the circuit of a make and break igniter, or to the coil of the high tension system, low tension igniters and dynamos being quite frequently used for the jump spark system.

There are two forms of low tension magneto systems, one in which the armature of the magneto revolves continually, and current is taken off the same way as it would from a dynamo for any other purpose, as for example for electric lighting systems. In the other form, the armature of the magneto is revolved through about one-quarter a circle by releasing a spring attached to a lever on the armature, and at the same time, the igniter points in the hammer-break type of igniter are separated.

The latter system is illustrated in Fig. 23, in which, at *I*, is shown the magneto at the left with the permanent magnets *b*, and a movable shield c which revolves between

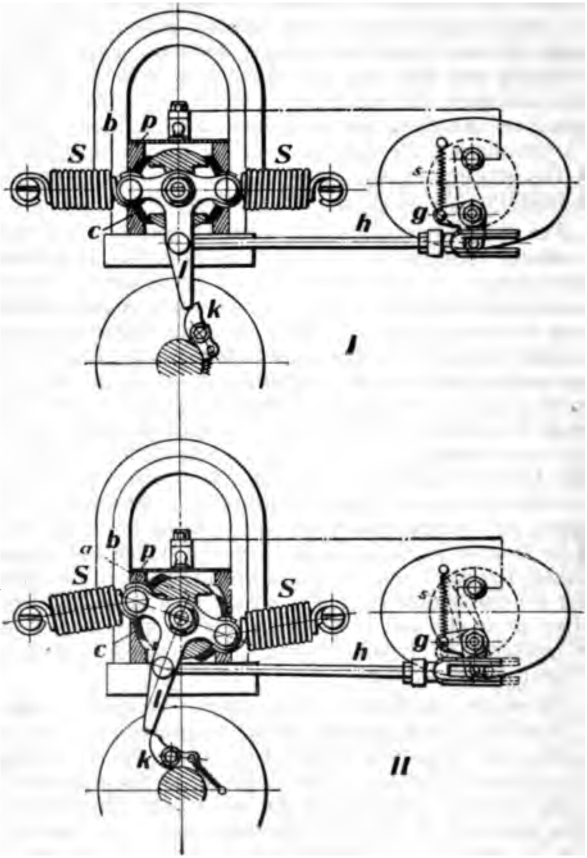

FIG. 23.

the armature and the magnet poles *p*. The rod *h* in the position as shown separates the igniter points indicated by dotted lines. As the proper time for ignition approaches, a dog *k* on the cam-shaft comes in contact with the lever *l*, which is attached to the shields *c*.

In diagram II, the cam *k* has swung the lower end of the lever *l* to the left, and with it the shields *c* against the resistance of the large springs *S*, *S*, and at the same time the rod *h* has been drawn to the left allowing the spring *s* to draw the movable electrode on the inside of the engine. The moment after the lever *l* slips off the end of the dog *k*, the springs *S*, *S*, pull the lever back into the position shown in *I*, and the shield revolves to the left, generating a momentary current by breaking the lines of magnetic force passing from the poles *p*, through the armature *a*. This answers the same purpose as moving the armature *a* through the same arc, as already referred to.

The maximum current is generated just as the point of the shield passes the corner of the armature core, and at that moment, the rod *h* strikes the point of the lower end of the rocker arm *g*, separating the igniter points on the inside of the engine. The two, working together, cause the break to be made just at the point of the highest electrical pressure, and an intense spark results. This system which is modified more or less according to the maker, is in extensive use on large stationary engines.

CHAPTER X.

JUMP SPARK IGNITION, HIGH TENSION COILS AND MAGNETOS.

In the jump-spark system of ignition a current of high pressure is used which will jump across a gap without bringing the points in contact as in the primary system. In a coil of wire which is around a magnet the pressure produced by altering the magnetic field is in proportion to the speed of the change, the variation in the strength of the field and the number of turns of the conductor surrounding the core of the magnet. If instead of a small number of turns of coarse wire, as in the ordinary spark coil, a very large number of fine wire turns should be substituted, the spark would have a greatly increased pressure. As it would be impractical to carry the current of an electric battery through a large number of very fine turns, two coils are used with a jump-spark coil. These are a winding of a few turns of coarse wire surrounding the core, and superimposed upon this winding is a secondary winding consisting of a great number of turns of very fine wire. When the circuit is closed and then broken through the coarse winding, known as the primary coil, a current of very high pressure is induced in the secondary winding. If the terminals of the secondary winding are close together or are connected to a specially devised plug having two terminals which approach each other, but are not in actual contact, a spark will jump between them.

In Fig. 24, is shown a diagram of the jump-spark system. In the figure *a*, is the iron core made up of a bundle of soft iron wires. Surrounding this is the primary winding *p*, while about the primary winding is a secondary winding, *s*, connected with the spark plug. The spark plug consists of a brass or steel body, *m*, which forms one side of the circuit. In the center of the plug is a wire insulated from the body by either mica, porcelain or other material capable of resisting the high pressure

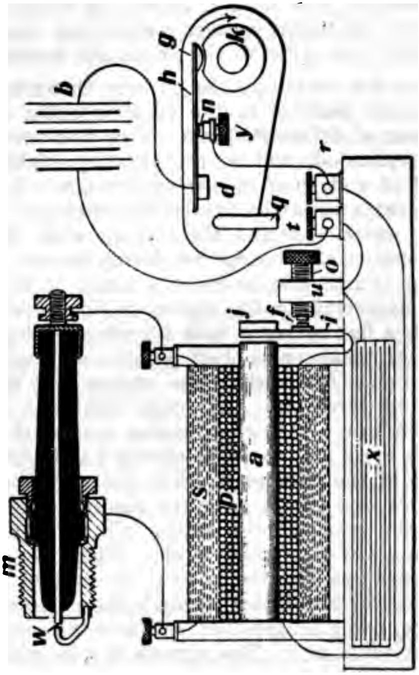

FIG. 24.

of the secondary. The other terminal is usually a nickel wire projecting from the body *m*, which approaches but does not quite touch the insulated electrode, forming the gap at *w*. This plug is screwed though the wall of the cylinder or the valve box of the engine so that the gap *w* is within the compression space. The primary circuit breaker or vibrator consists of a flat spring *i*, to which is fastened a piece of soft iron, *j*, called the armature. At *f* are a pair of platinum contact points, one fastened to the spring *i* and the other to the adjusting screw *o*.

The device for switching the current through the coil at just the right moment in the cycle is called a timer. Quite a number of different forms are in use, some simple and some complicated, and in the cheaper engines, generally consist of a strip of copper or brass held by spring pressure against a collar of insulating material, carrying a metal bar which connects the circuit with the metal work of the engine, this connection being termed, grounding it. In the illustration is shown a timer of the motorcycle type. Referring to the figure, a fiber body *d* has fastened to it a flat spring *h* with a knob *g* at the further end. A pair of platinum contact points are located at *n*, one of them being fastened to the spring and the other to an adjusting screw *y*. A notched cam *k* is attached on the cam-shaft on a four-cycle engine and on the crankshaft on a two-cycle engine. The spring *h* is in connection with one pole of the battery *b*, while the other pole is in connection with one of the primary binding posts of the coil. The other binding post of the coil is connected to the adjusting screw *y* of the switch. When the notch in the cam *k* comes opposite the knob *g* the contact points *n* meet, and the circuit is closed through the primary of the coil. The core, being magnetized, attracts the armature *j*, breaking the circuit at *f*. The armature *j* is immediately released, allowing the contacts *f* to come together, and the spring continues to vibrate so long as the circuit is closed at *n*.

To the fiber back *d* of the switch is attached a rod *q* by means of which the switch may be rotated about the cam-shaft, changing the time in the rotation of the engine when the circuit will be closed at *n*. Rotating the switch in the same direction as the cam revolves will make the switch operate later and reduce the lead of the ignition, while if the switch is rotated in the direction opposite to the motion of the cam the lead of the ignition will be advanced.

At *x* in the figure is shown what is known as an electric condenser. The condenser usually consists of a number of sheets of tinfoil between which are placed sheets of oiled paper. Every other sheet of tinfoil is in metallic contact, but insulated from the tinfoil in between. The other sheets of tinfoil are also placed in metallic contact. Hence there are in the condenser two sets of these metallic sheets, one set insulated from the other, forming the two sides of the condenser. One side of this condenser is connected with the spring *i* and the other to the adjusting screw *o*. In other words, it is in shunt or parallel with gap *f*. The condenser has the effect of reducing the volume of the spark at *f*, which, if very great, prevents the spark occurring at the gap of the spark plug.

It will be seen that in a coil of this kind a series of sparks will be produced by the vibrator each time contact is made at *n*. In some types of this coil the vibrator is omitted and but one spark is made whenever the knob *g* drops into the notch in the cam, or more correctly when the knob *g* is raised from the notch, breaking the circuit at *n*. As explained in Chapter IX, it takes a definite time for the core of the coil to reach full saturation. Therefore the change of the magnetic field when contact is made at *n* is so slow, comparatively speaking, as to produce no spark at *w*, the spark being made when the circuit is broken. Sometimes the switch is so arranged for a non-vibrator coil that a spring *h* is set in the vibration, this having the same effect as the vibrator spring *i*, forming a series of sparks at *w*. It should be noted that in this system, the secondary coil *s* becomes practically a source of current for the secondary circuit, the energy in which is generated by the variations in the lines of force caused by the pulsating current in the primary circuit.

In coils for ignition all parts with the exception of the vibrator and the binding post, are placed within a case made of wood or other material, and the space between the parts filled with an insulating compound, usually paraffine or some other wax mixture in order to protect the parts from moisture. Coils of what are generally known as the "*dashboard* type" are made up in units, each unit in a thin wooden case, and these are placed side by side in a heavier wooden box containing as many units as there are cylinders in the motor for which it is designed. The units are made with spring contacts and are held in the outer case by spring latches and the binding posts are arranged on the bottom of the outer box. This plan permits one unit to be removed in the event of anything going wrong with it, and replacing it with another.

Usually each unit has its own vibrator, although sometimes one vibrator, called a master vibrator, is so connected that it breaks the current for each one of the units in turn, and this vibrator is arranged with its own electro-magnet, wound with only the primary winding. This has the advantage that the vibrator is the same for each coil, and therefore the ignition is more perfectly timed, but practice seems to favor a vibrator for each individual unit, because if the master vibrator should go wrong, it disarranges the entire system, while with the single vibrator for each coil, one vibrator simply affects the cylinder to which the particular coil is connected.

In some of the later systems only one coil is employed for any number of cylinders, the secondary or high tension circuit being switched from one cylinder to another by a high tension timer or as it is usually called, a distributor. A distributor system when used with the vibrator coil requires a coil and a vibrator that are both very fast and for a high speed engine, it is not altogether satisfactory. In the first place it puts all the work on one vibrator, and secondly because the coil is in practically continuous use, the core, if not ventilated, is apt to become overheated. There is still another objection and one that is not generally appreciated. A vibrator coil produces a series of sparks that appear to break down the insulating properties of the slight air gap between the distributing finger and the contacts from which the wires lead to the spark plugs, and the spark, instead of going to its proper plug, is deflected where it is not wanted. The author has seen, in a distributor used with the vibrator coil, the spark jumping across a space as wide as a half an inch when the gap between the distributor finger and the terminal of the spark coil wire was scarcely a sixty-fourth of an inch. For these reasons the most successful distributor systems are those used with a coil having no vibrator and in which

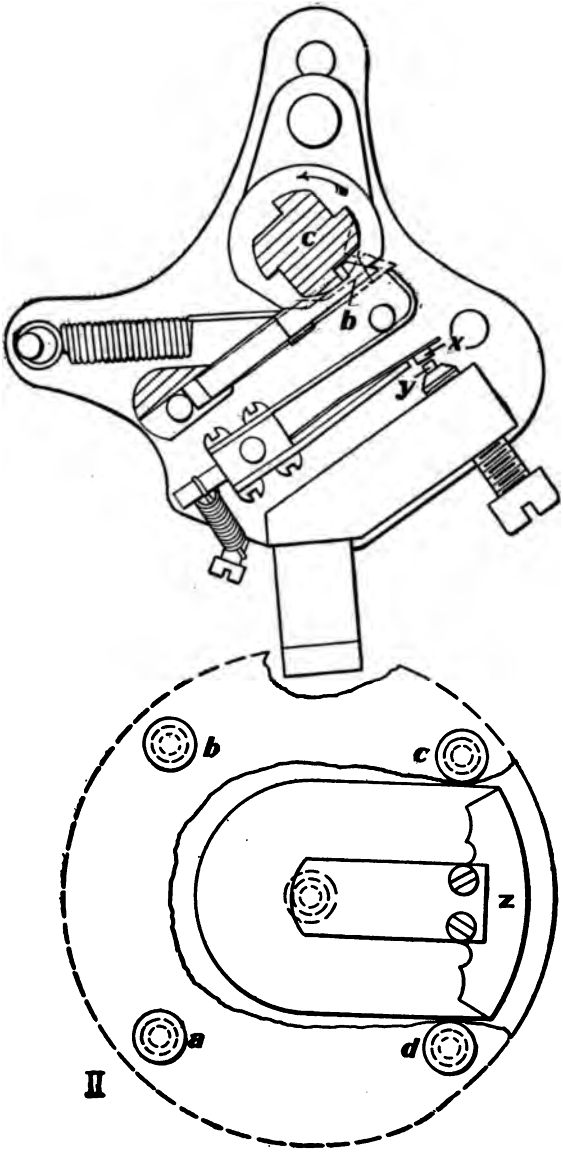

FIG. 25.

the spark is made by the quick separation of the timer points, which make and break a circuit of the primary winding. This overcomes all the objectionable features noted above and, there being but one spark produced for each ignition, there is no heating of the core of the coil, no vibrator to be overworked, and the single high tension spark does not have the same tendency to deflect from its proper path in the distributor.

There are two forms of single spark distributing systems, one in which the quick breaking of the primary circuit is produced mechanically, and another which is produced electrically. In the mechanical break, one of the contacts of the timer is so arranged that it touches but momentarily the other contact of the timer, by a sort of snapping motion. In the electric breaker, current is passed through an electro-magnet and at the same time is passed through the coil, and immediately the core of the electro-magnet is energized by the current passing around it and the circuit is broken just as described in Fig. 24, with this exception, that the current through the electro-magnet of the circuit breaker is not broken and the contacts are held apart until the timer on the engine opens the circuit.

The principle of the mechanical break referred to above is shown in Fig. 25. The small cam *c* is pinned or keyed to the timer shaft of the engine and turns in the direction shown by the arrow. As it revolves it catches the tiny hook on the end of the blade *b*, dragging it with it until it has reached the point shown by the dotted lines when the blade *b* slips off the point of the cam and sliding over the hump on the cam as shown, going a little farther from the center, and on rising, brings into contact for a moment only, the platinum point *x* on the blade, with the platinum point *y*, on the stationary electrode. The distributor portion is shown in Diagram II, the shaft carrying the contact finger *z*, which distributes to the points, *a*, *b*, *c*, and *d*, and thence to the various spark plugs. As the distributor shown is designed especially for a four cylinder engine, a spark is produced each time the tongue *b* slips off the point of the cam, while the contact finger on the distributor portion is opposite its corresponding distributor point *a*, *b*, *c*, or *d*. The duration of contact between *x* and *y* is approximately one two hundredth part of a second, and it is only while these points are in contact that the current is passing through the coil, but very little current is used in this system as compared with that in the vibrator coil. This system is not only very economical of battery, but if the cam *c* is carefully made, the time of the contact is very accurately distributed, and the contacts are made to occur exactly ninety degrees apart.

For starting on the spark a push button is used which closes the primary circuit and produces a spark when this push button is released. This system has one disadvantage, that, while it gives very satisfactory ignition when the motor is once started, the condition of the mixture must be exactly right or the motor is very difficult to start.

The single spark system using the electric break is shown in Figs. 26-I and 26-II. In I is shown the mechanism of the electrical break and the wiring diagram in connection therewith. An electro-magnet employed to open any form of switch or to operate any other electro-magnet or electrical device is called a *relay.* In the particular form of relay used on this device there are two windings on the magnet. One of these windings is of coarse wire which allows sufficient current to pass around the magnet to attract the armature controlling the switch or circuit breaker. The other winding is of very fine wire and permits the passage of just sufficient current to hold the armature of the relay so it will keep the switch open. This fine wire winding is called the "holding coil."

In the figure the circuit passing through the primary timer *T* is as follows. Suppose the battery switch *S* to be closed. (It is open in the figure.) Then current will pass from the ground *G*, through timer *T*, the battery *B*, the primary *P* of the spark coil, the coarse winding *c* of the relay, the relay switch *s* and the battery switch *S* to the ground *g*. The current in *c* energizes the magnet *m* and, attracting the iron lever *l*, opens the switch *s*. So long as timer *T* is on contact, a small amount of current will flow through the holding coil *h*. This current is so minute that the break of the circuit at *s* is sufficient to cause a high tension current in the secondary *H* of the coil. This high tension current is lead by the distributor *D* to one of the plugs *p*.

For starting, the button *b* is pushed in to open the switch *R*, thus disconnecting the holding coil from the ground. Hence when the spring q closes the relay switch

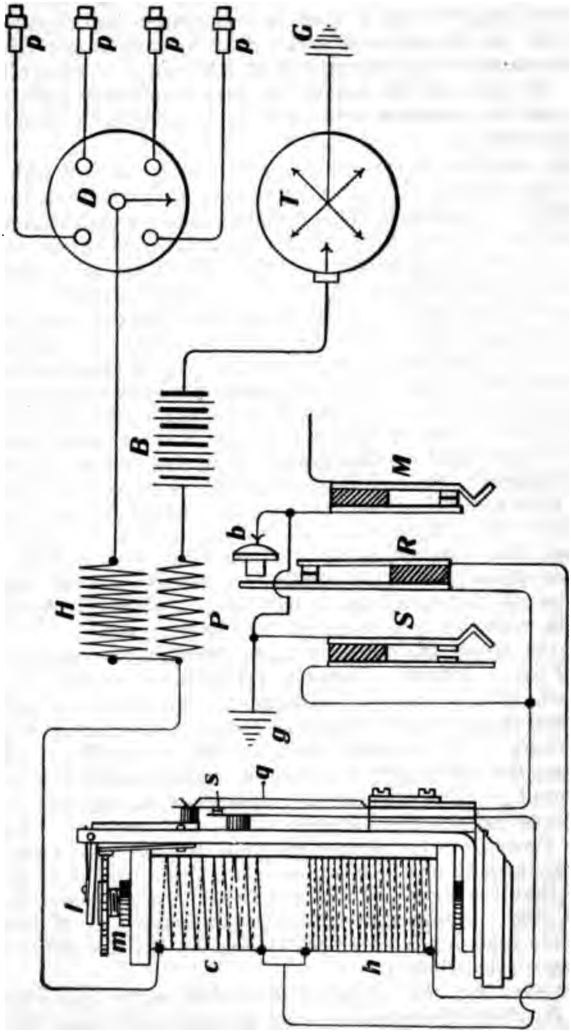

FIG. 26-I

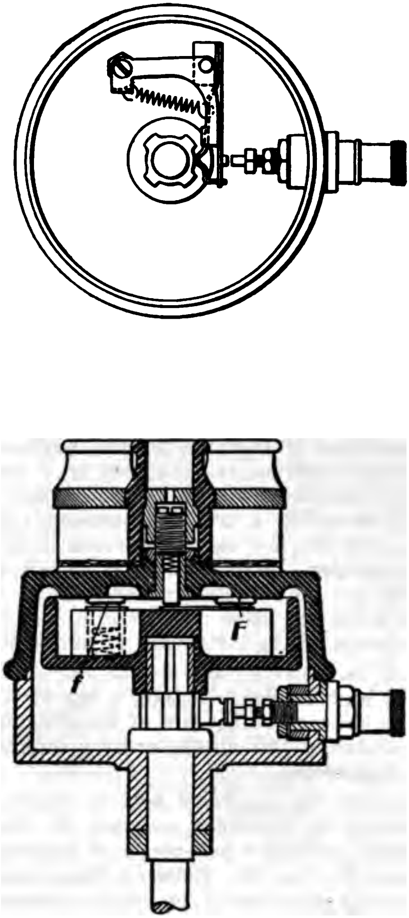

Fig. 26-11.

s it acts as a vibrator. After the engine has started, the button *b* is retracted and the switch *R* is closed. The switch *M* is provided for use with a high tension magneto, in case the latter is used on the same motor, as an independent system of ignition.

Another unique feature of this system is the arrangement whereby the spark on the distributor is prevented from going to the wrong circuit, and thence to the wrong spark plug. This is accomplished by having a grounded distributor finger *f* (diagram II) in advance of the regular distributor finger *F*, so that in case there is a tendency of the spark jumping to a cylinder in which the piston has not reached the proper point in the stroke for ignition, it will go to the ground and not to the spark plug. The distributor fingers rub against the contact points, from which the wires lead to the various spark plugs of the system.

The high tension magneto combines in one unit the functions of the source of current, the high tension or jump-spark coil and the timer and distributor. While there is a variation of the high tension magneto in which the high tension current is generated in a non-vibrator coil which is separate from the magneto itself, this form can scarcely be called a true high-tension type for the reason that only the low tension current is generated in the magneto proper. As this style of so-called high-tension magneto has the circuit breaker and the distributor arranged in much the same manner as the self-contained type a description of the latter will be ample for both.

In Fig. 27 is shown a cross-section and an end view of a well-known high-tension magneto, a specific form being shown in this instance on account of the difficulty of illustrating a machine of this character by a diagram so that it would be understood.

In the figure the magnetic field is comprised of the three horse-shoe or U-shaped magnets *M*, the poles of which are faced with the pole-pieces *P* illustrated in the small Diagram II, Fig. 28. Between these pole-pieces revolves the shuttle armature *A*, on which is wound both the primary and the secondary coil. (See Fig. 28.) The cutting of the lines of force generates a current in both the high and low tension coils or windings, but the extreme high tension necessary is secured by breaking the

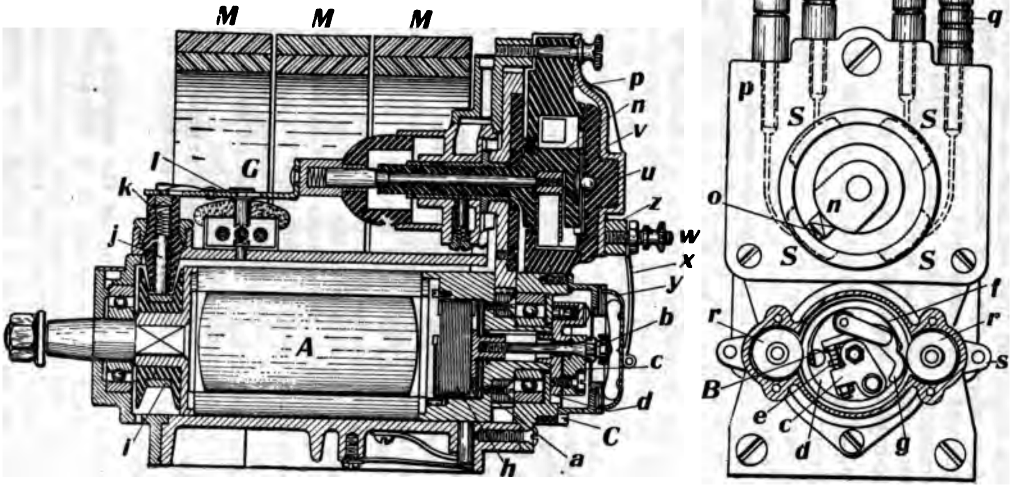

FIG. 27.

primary circuit at just the time in the revolution of the armature when a spark is desired at the plug. As there is a current wave generated twice in each revolution of the armature, the breaker must act twice each revolution and for a four cylinder motor, the distributor must rotate at one-half the speed of the armature.

Turning to the longitudinal section of the magneto, the circuits may be easily followed with the aid of the ensuing description. In the first place it should be noted that in common with the majority of ignition systems one side of the circuit is through the metal work of the motor, *i. e.*, it is grounded. One end of the primary circuit is grounded by being attached to the armature core. One end of the secondary winding is attached to the opposite end of the primary and forms a continuation of the primary. Thus, the secondary is grounded on the armature core through the primary, while from the opposite end of the secondary the high-tension current is led through the distributor to the spark plugs. This path can be followed quite easily in Fig. 28.

In Fig. 27, the end of the primary where it is joined to the secondary is connected to the brass plate *a*, whence it passes through the screw *b* and the block *c* to the platinum screw *e*. Parts *b* and *c* are insulated from the breaker disk *d*, which is connected to the armature core and through this to the grounded end of the primary. Attached to the disk *d* is the block *c* carrying the contact screw *e*. During the greater part of the revolution of the armature the spring *f* holds the lever *g* with the screw *B* in contact with *e*, thus short-circuiting the primary. Twice in each revolution of the armature, the end of the lever *g* comes in contact with fiber roller *r*, drawing the contact screw *B* away from *e* and the primary circuit is opened, and a high-tension current is generated in the secondary winding.

The ungrounded end of the secondary is attached to the slip-ring *i*, and thence it is taken off by the carbon brush *j*, which is insulated from the magneto frame by the brush holder *k*. From the brush the secondary current passes to the bridge *l* through the contact carbon, the rotor *n* of the distributor, and the distributing carbon *o* to the distributor segments *S*. From these segments the

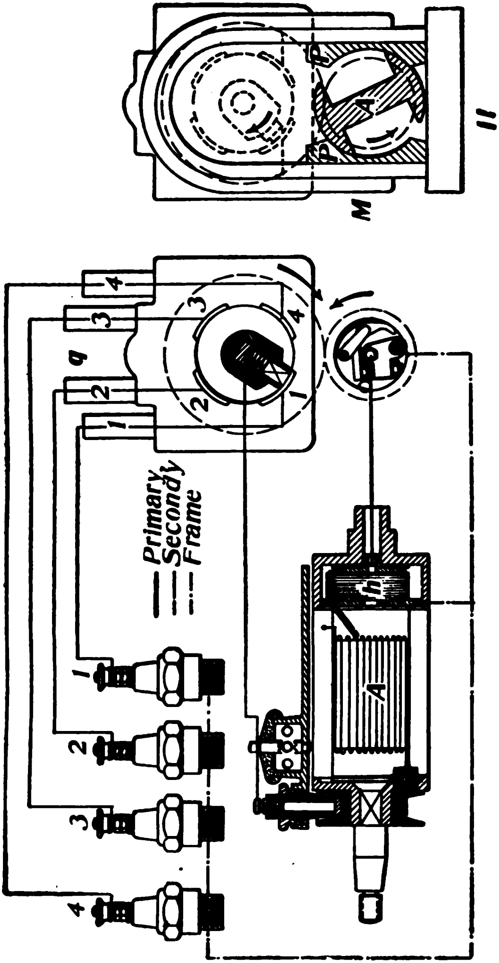

FIG. 28.

current passes through the contact plugs *q* and the high tension wires to the spark plugs.

In order to reduce the flash at the contact breaker between screws *e* and *B* to a minimum, a condenser *h*, is connected in parallel with this gap. At *w* is a binding post for short-circuiting the primary. This binding post is connected through the spring *x* which holds the brass cap *y* in place to the end of the screw *b*. From the binding post *w* a wire is led to the magneto switch and thence to a ground on the motor. Whenever this switch is closed, the breaker cannot open the primary circuit, and no current is generated in the secondary. At *G* is a safety gap, the lower pole of which is grounded on the frame of the magneto and the upper connected to the bridge *l*. If for any reason a cable is disconnected from a spark plug, the spark will jump this gap and save the magneto from injury.

In Fig. 28 is shown a diagram of the wiring and connections for this magneto. The primary winding is given as a heavy line, the secondary is shown by fine lines and the ground or frame by dotted lines. The different parts are lettered the same as in Fig. 27. It should be noted that the order of firing as shown is very seldom used in a motor, and the plugs *q* must be connected to the spark plugs in the order the cylinders should fire.

For a four cylinder motor of the four-cycle type this magneto should be run at crank-shaft speed, and on a two-cycle at twice crank-shaft speed. The three-cylinder magneto should be driven at two-thirds crank-shaft speed on a four and one and one-half crank-shaft speed on a two-cycle. The two-cylinder takes half shaft speed on the four and the same speed as the shaft on the two-cycle. The six-cylinder requires to be driven at one and one-half shaft speed on the four and three times the speed on the two-cycle. The ratio of the distributor speed on the magneto is according to the number of cylinders on the motor, for example, it is one to two on the four-cylinder magneto.

There are now on the market magnetos so designed that four sparks are produced at every revolution of the armature and for these the ratio of the speed of the magneto to that of the motor is just half those given in the last paragraph. There is quite an advantage in this, especially for a two-cycle high speed motor. For example, on a six-cylinder two-cycle when the magneto is driven at three times shaft speed, and the motor is running 1,200 r. p. m., the magneto is running 3,600 r. p. m.

The reader has undoubtedly noted that the high-tension magneto must be driven by gears or other means that will make the speed of the magneto a fixed ratio to that of the motor, and therefore bring the spark at the same point in the stroke each time.

Methods of spark advance for magnetos will be fully discussed in Chapter XXIII.

CHAPTER XI.

MISCELLANEOUS IGNITION SYSTEMS.

The earlier engines of the four-cycle type employed flame ignition. The principle of this form of ignition is well shown in Barnett's igniting cock, Fig 29. In this method two gas jets are necessary, one the flame *f* which is employed for the ignition of the charge, and the other the flame *F* for relighting *f* when blown out by the flame within the cylinder. The plug valve *A* is shown in the proper position for relighting the flame *f*. At the right moment, the valve makes a quarter turn so that the opening in the plug is opposite the opening in the cylinder, as shown by the dotted lines. The resulting explosion extinguishes the flame *f*. The valve on returning to the position shown in the figure allows the gas to rush out and ignite from the flame *F*.

After the flame, the hot tube igniter came into use. In this form of ignition, a tube closed at one end is arranged with its open end connecting with the compression space within the cylinder. Two methods of timing are employed. In one, the tube is opened to the compression space by means of a valve. In the other and more common method, the tube is left open to the compression space at all times and the ignition timed by the distance of the heated portion of the tube from the mouth. After exhaust, a small portion of the burned gas remains in the tube, and during compression this is driven back by the new mixture to such a point that will allow the fresh charge to come in contact with the hot spot on the tube and ignite.

An example of a hot tube igniter with an adjustable burner, for altering the time of ignition is shown in Fig. 30. The tube is shown at *t*, and should be made of some refractory material such as nickel alloy or porcelain. The fresh mixture is driven by the compression into the port *p*, whence it passes to the tube. The tube is heated by means of the Bunsen burner *b*, the flame entirely surrounding the tube, being confined by means of the chimney *c*. The chim-

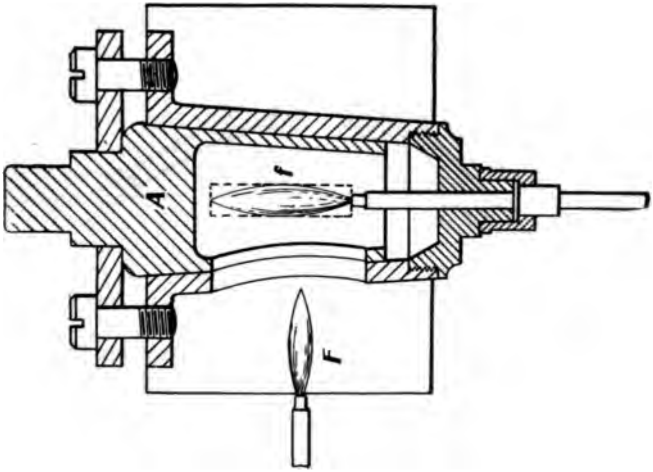

FIG. 29.

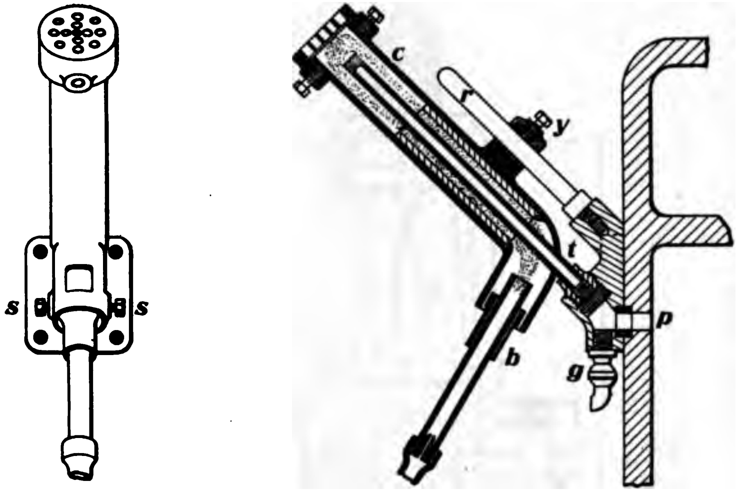

FIG. 30.

ney is lined with asbestos to prevent the heat from escaping through the walls of the chimney. Minute adjustments of the flame are obtained by swinging the burner about the set-screws *s*, and larger adjustments by loosening the screw *y* and sliding the chimney along the rod *r*. The pet-cock *g* is provided for blowing the soot from the tube.

Both the flame and the tube igniter have the objections that they require additional fuel, are sensitive to drafts and other currents of air and increase the fire hazard.

In quite a few oil engines, ignition is obtained by means of a division of the compression space which is not water-jacketed. Into this space the oil is pumped in a jet and, being driven against the hot wall of this compartment or bulb, is vaporized. The necessary air is drawn into the main part of the cylinder and the compression finally drives it into the heated chamber. When sufficient air has entered the bulb to form an explosive mixture within the bulb, ignition takes place. In the Diesel cycle we have a similar form of ignition, the fuel being separated from the air until the moment of ignition and fired by the heat of compression. See Chapter I.

What might be called a modified flame ignition has lately been successfully employed, in which a portion of the charge is retained in a compartment of the compression space controlled by a valve, which is opened at the proper time to ignite the following charge. The particular motor in question is a two-cycle, and although the author has not had an opportunity to see the engine he is assured that it gives a very satisfactory performance. It is necessary to provide means for securing the first ignition, after which the ignition is automatic.

Catalytic ignition (literally "contact" ignition) is based on the following principle. If platinum in the form of exceedingly fine wire or platinum sponge be brought into contact with an inflammable gas or vapor it will be brought to incandescence and finally ignite the gas. Under compression this is increased. Based upon this principle, modified forms of catalytic ignition have been tried with but moderate success. In one a platinum grid was inserted in a slide and once heated from the combustion or explosion retained a sufficiently high temperature to ignite a succeeding charge when brought in front of an opening in the compression space at the right moment. In anoth form, a coil of wire was kept at incandescence by passing a current through it and brought into contact with the charge at the right moment. Neither of these two systems proved commercially successful and true catalytic ignition did not, so far as I know, get beyond the mere suggestion.

If we should so arrange the vibrator of a spark coil that it could be inserted within the cylinder of a gas engine it would readily give a spark of sufficient volume to ignite the charge. Based on this principle we have various forms of magnetic spark plugs, one at least of which has proven very successful in practice, although others have shown very poor lasting qualities.

In order to overcome the effect of salt water dashing upon the high tension wiring of the jump-spark system various plans have been tried. One of the best of these is, perhaps, the combined coil and plug in which no part of the high tension circuit is exposed to the elements. These combined coils and plugs are known under various trade names and usually called "Ignitors." The spark plug forming part of the spark coil is connected with the high tension winding entirely within the casing and therefore all wiring carries low tension current, which is not affected by moisture.

A cross-section of one of these devices is shown in Fig. 31. The primary coil is shown at *p* and the secondary at *s*. Usually these coils are operated from the ordinary timer through a master vibrator as shown in the wiring diagram in the figure.

If across the terminals of secondary wiring of a spark-coil we bridge a condenser, the length of the spark is reduced, but its volume is increased, and the spark is altered from a violet tinge to a bluish white. This spark is due to the high frequency discharge of the condenser. If, from the terminals of a coil connected in this way, we lead wires to a spark plug there is secured a spark of great persistence that defies soot, water and oil. This system or modifications of it has been developed in England by Sir Oliver Lodge as well as by an American manufacturer.

The diagram of the Lodge ignition system is shown in Fig. 32. Current is applied from an eight volt storage battery to the low tension terminals *LL* of the spark coil. n the high tension side the condenser *C* is bridged across

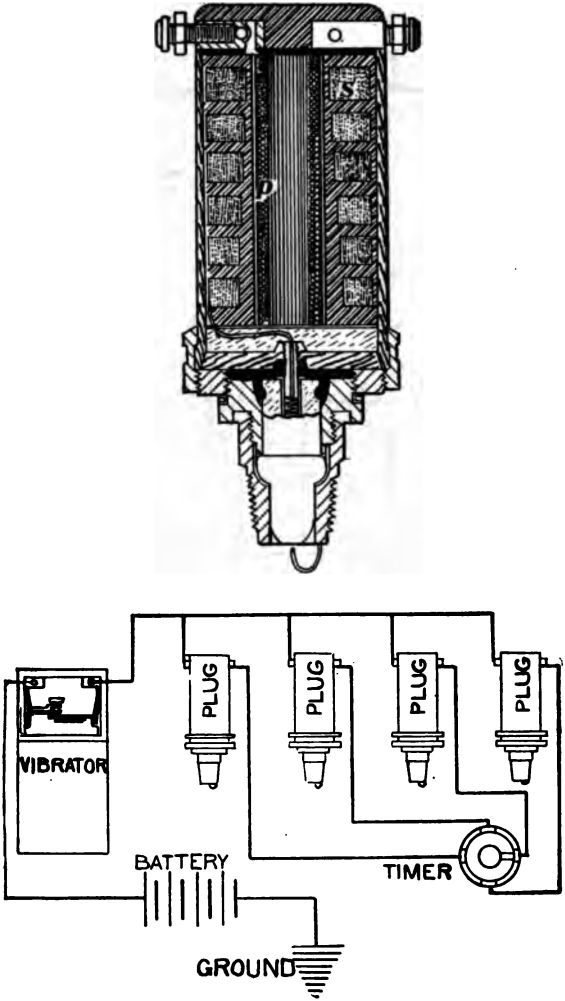

FIG. 31.

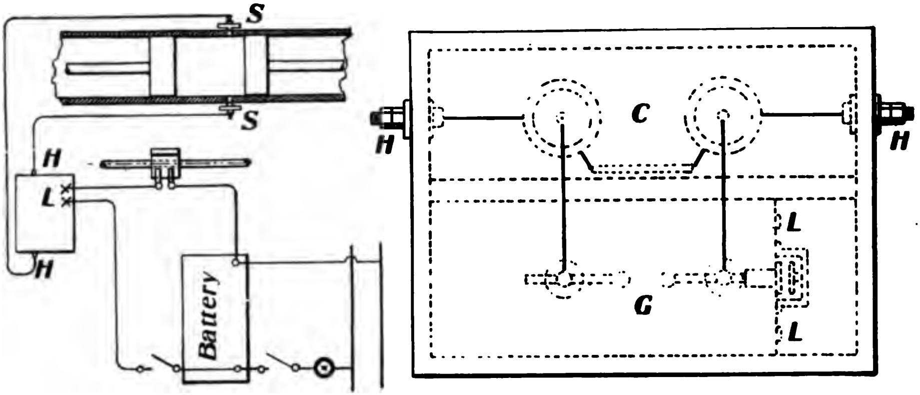

FIG. 32.

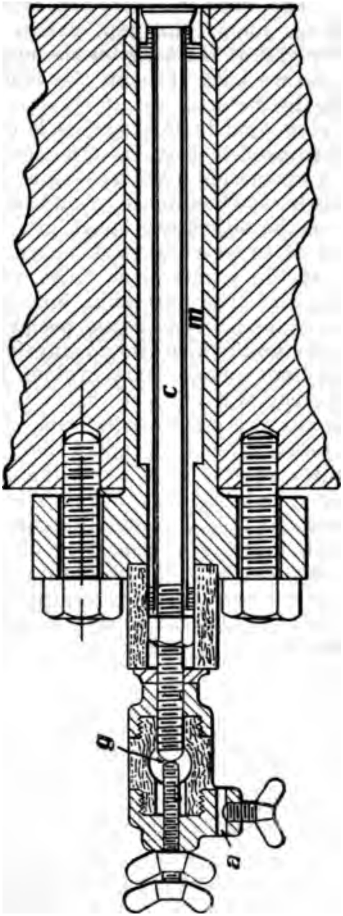

FIG. 33.

the spark gap and from the high tension terminals *H* the wires lead to the two spark plugs *S* on opposite sides of the cylinder. The current enters the central electrode of one spark plug, jumps the gap, passes through the engine, jumps the gap of the plug on the opposite side of the engine and thence back through the central electrode of the other plug to the coil.

The spark plug used in this system is one of special design and is illustrated in Fig. 33. The central electrode *c* terminates in a mushroom head making an annular gap. The electrode *c* is surrounded by mica and terminates outside the engine in the adjustable spark gap *g*. The purpose of this gap is to prevent weak sparks creeping to the spark plug at the wrong time. The wire from the coil is connected at *a*. The particular form of gap inside the cylinder leaves no point to cause premature ignition and the large surface, which would involve danger of leakage in the ordinary jump-spark system, has no effect on the high frequency system.

This system has two advantages. One is the absolute immunity to ordinary causes of spark plug trouble and the other the entire absence of shock to the operator if he touches the plug. Outside of a burning sensation if a finger is held close to the plug and the spark allowed to creep to the hand, there is no sensation on touching the plug. Its only drawback is the fact that it requires a greater volume of current than the ordinary jump-spark system. This is not of sufficient moment to prevent its use and it is a system that is full of promise.

CHAPTER XII.

GAS ENGINE GOVERNORS.

For regulating the speed of a gas engine there are two general methods employed. One, the *intermittent impulse* system, in which the full charge of fuel and air is taken into the cylinder for every impulse, and as the load decreases, one or more impulses are omitted. The other method, the *variable impulse* system, in which the mean effective pressure within the cylinder is reduced as the load decreases.

The intermittent impulse system, commonly known as the "hit-or-miss," does not give close regulation, but the variations of speed, especially with a comparatively heavy flywheel, are not too great for the ordinary class of machinery. It has the advantage of simplicity, and also of economy for each charge is burned under the conditions favoring the highest economy, that is, with a full charge and high compression. For driving electric generators, this method has been employed with fair success by using extremely heavy flywheels, but for close regulation, say, within two per cent or less, it is not satisfactory, and for such delicate work as running alternators in parallel, it is practically useless.

The variable impulse system may be divided into two sub-methods, one in which the quality of the mixture, that is the proportion of the charge of fuel and air, is kept constant and the quantity of the mixture is reduced as the load goes off. Another method is that in which the quantity of the mixture is kept constant, and the quality is altered, either by changing the proportions of the mixture, reducing the amount of fuel in the charge, or by admitting burned gases from a previous charge to dilute the mixture.

In the *hit-or-miss* system there are four methods of regulation, first, keeping the fuel valve closed; second, holding the exhaust valve open, a method common where the automatic inlet valve is employed; third, holding the intake valve closed, and fourth, switching off the ignition current to produce the idle stroke. In the first method, the intake and the exhaust valve operate normally and a charge of air without fuel is drawn into the cylinder, compressed, expanded and exhausted. In the second method, holding the exhaust valve open, exhaust gases from a previous charge are drawn into the cylinder from the exhaust passages and blown out again, but not compressed, as the exhaust valve remains open. In this method, especially when gasoline is the fuel employed, a lever, operated by the governor mechanism, holds the intake valve to its seat. When the inlet valve is held closed and the exhaust valve is operated normally, the piston draws a vacuum during the suction stroke. In the fourth system, a charge is drawn in, but not ignited, and is exhausted through the exhaust valve. Of the four systems, the two first are those most used, and that in which the fuel valve is kept closed, allows the scavenging of the cylinder on a charge of pure air during the idle stroke. The second method, holding the exhaust valve open, appears to be the most frequently used, especially in American practice. The third method requires powerful springs to hold the inlet and exhaust valves to their seats and it is for this reason not desirable. The fourth method, omitting the ignition of the charge, is extremely wasteful of fuel, and the author knows of but one or two instances in which it was employed, or even suggested.

In the *quantity method* of the variable impulse system governing is accomplished in several ways. Plain throttling or choking of the charge by means of a throttle valve attached to the governor is perhaps the most common. This method requires a small though appreciable amount of work to draw the charge through the restricted passage, for which work there is no return. As the quantity is reduced, the pressure after compression is reduced a corresponding amount, with consequent loss of economy. There is a small gain in economy due to a small increase of expansion, but it is scarcely enough to be of moment.

The variable cut-off method of the variable impulse system consists in holding the inlet valve wide open during a portion of the suction stroke, and suddenly closing it at a point determined by the governor just in the same manner as on the admission valve of a Corliss steam engine. *Following* the cut-off, the pressure within the cylinder falls below that of the atmosphere, and returns to atmospheric when it reaches approximately the point of cut-off on the return or compression stroke. The compression is lowered according as the cut-off is earlier, but the expansion is carried much further than in an engine of the ordinary four-cycle type. The increase in expansion balances to a considerable extent the loss of economy due to lower compression and the work of lowering the pressure in the cylinder after cut-off is regained during the first part of the compression stroke. This is the governing system employed on the Sergeant engine and is illustrated most clearly by means of the diagrams in Fig. 34.

There has been employed still another plan of the variable impulse system, holding the inlet valve closed for the early part of the suction stroke and opening it later as the power requirements became lower. The author does not recall its use on motors of the stationary type, but it was employed, for a time, at least, on the motor of a well-known automobile. Its chief recommendation appears to be the sharp draft of the air through the carbureter at low loads.

Forcing a portion of the charge back into the intake passages is a fourth method, and is accomplished by closing the inlet valve late in the compression stroke, varying the point of closure according to the load requirements.

The *quality method* of the variable impulse system has the advantage of retaining the same compression pressure on light loads as on full load. It has the disadvantage that weak mixtures are both difficult to ignite and slow burning. It is used to some extent in combination with the quantity method, and even with the intermittent system.

Regulation by the quality method may be accomplished by varying the lift of the fuel valve, or decreasing the time during which it may remain open. In the earlier Otto electric light engine, which employed this system, the final regulation for extremely light loads was by cutting off the fuel and omitting the impulse entirely.

Drawing back a portion of the exhaust gases by closing the exhaust valve late, cuts down the quality of the charge, in a way. It may be considered, however, that this plan belongs as much to quantity regulation, although the com-

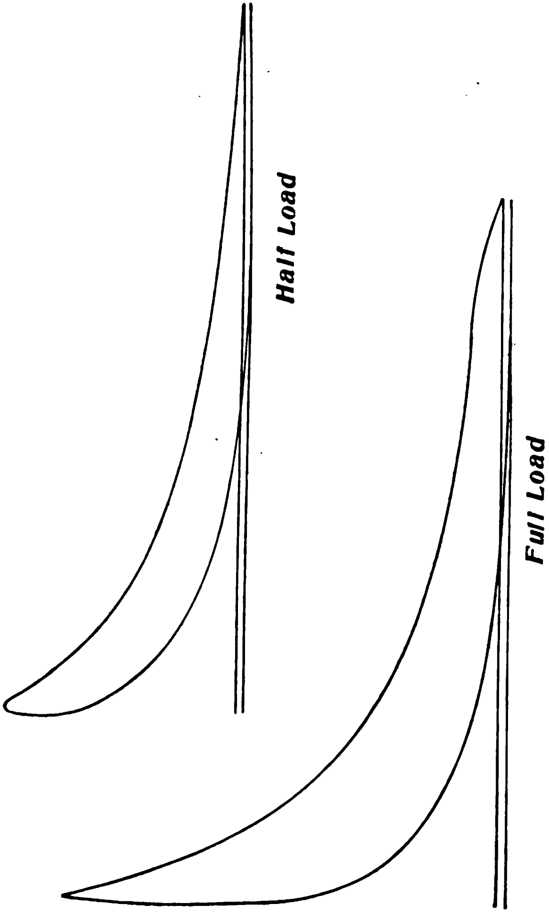

FIG. 34.

pression pressure is not reduced by this system. It would appear that this method is virtually what takes place in the regulation of the two-cycle motor in which the fresh charge is admitted from the pump or crank-case, through a piston controlled port. While ostensibly the method of regulation is by means of a throttle either in the inlet passage to the crank-case or pump or in the passage leading to the cylinder, a portion of the exhaust from a previous explosion will remain when there is not sufficient of the new charge to drive out all of the old.

In the larger engines of the two-cycle type served by separate air and fuel pumps, quality regulation is employed by varying the quantity of fuel delivered by the pump. As these engines receive a scavenging charge of air, the mixture at low loads is one weak in gas.

Some builders prefer to regulate by varying the time of the ignition. This is, however, only moderately effective and is wasteful of fuel at moderate or small loads. Running with late ignition will heat up the exhaust passages because the completion of the combustion of the charge is delayed until after exhausting.

Variation in the timing of the ignition should be considered for the reason that mixtures that are weak in fuel or not compressed as much as at full load burn more slowly than rich or fully compressed charges. This is best practice but not often followed. It gives much better economy at partial loads to advance the ignition as the load decreases.

In the regulaton of the smaller two-cycle engines using base compression, if the throttle is closed with the ignition fully advanced, the motor will "four-cycle," (explode at every other revolution) and run jerkily. It is therefore necessary in motors of this type to retard the ignition in order to secure satisfactory regulation at no load and at very light loads.

It is obvious that any engine may be regulated by the use of two or more of the above plans in combination. Engines of different designs require different treatment, and experience, your own and that of others, is the best guide to the choice of the proper method to employ for any particular design of engine. The forms of mechanism employed for speed regulation will be taken up fully in Part II.

CHAPTER XIII.

GAS ENGINE STARTERS.

Engines of moderate speed, larger than eight or ten horsepower per cylinder are difficult to start by turning the flywheel by hand, and, in order to lessen the amount of labor involved, larger engines than the above are usually supplied with some special means of starting. One method is to use some device by means of which an impulse may be given to the piston when the engine is at rest, that will be strong enough to propel the engine for three or more revolutions and until it can take up its cycle and receive an impulse in the regular way. Another is to provide exterior means for turning the flywheel, either by power or by gearing. The methods in use for this purpose are so numerous that but a few examples may be illustrated in a volume of this nature.

The first method to suggest itself to an engineer would obviously be to apply pressure to the engine for one or more strokes from an exterior source, as from a steam boiler or from a tank of compressed air. In fact the compressed air starter is a most convenient form. It is very simple and consists of a small air pump and a storage tank for the air. A pipe connects the tank to the cylinder of the engine, and this pipe is supplied with a two-way plug-valve. The tank is filled with air, at a pressure of 40 to 200 pounds, when the engine is running, the pump being operated by means of a belt from the line shaft or other convenient point. When it is desired to start the engine the flywheel is turned until the cam-shaft is in the proper position for the expansion stroke, and until the crank-pin is just above the back-center. The air-valve is then opened wide, and closed again as soon as the piston nears the end of the stroke. If the impulse is not strong enough to give the engine a sufficient start it should be repeated at the next revolution but one. Two impulses with the air are usually more than enough for the purpose. After the engine is under way, the fuel should be turned on as directed in the chapter on *Starting and Control.* If the engine has more than one cylinder, one or more cylinders may be used to start while the remaining cylinders are taking up the cycle. Occasionally, the engine is fitted with a special valve mechanism, for the purpose of starting with compressed air, so arranged that an impulse is given to the piston by the air at every revolution until the speed is sufficient to take up the cycle.

To start an engine of medium size which would be too hard to turn rapidly, as in the case of an automobile motor, a plan quite frequently employed is to pull the engine over until it is just beginning the compression stroke, and either open the gas valve slightly to draw in a charge of gas, or prime the cylinders with gasoline. The engine is turned over until the suction stroke is nearly completed, and then the flywheel is pulled smartly backward to compress the mixture, and just before the rebound of the piston the igniter is snapped and the charge fired. This usually gives an explosion of sufficient strength to carry the engine through two revolutions.

In some cases, instead of snapping the electric igniter, a match is used in some such device as illustrated in Fig. 35. In this, the match *m* is held in place by the valve *b* and ignited by a smart blow on the plunger *A*. Opening the valve after the engine is started will blow the match out.

When the jump-spark is employed, a switch which will close the primary circuit, is all that is necessary. In other engines the flywheel is turned until the cam-shaft is at the proper position for the beginning of the suction stroke, and the crank-pin is on the inner center. A charge of fuel and air is then pumped into the cylinder by hand. The pressure within the cylinder can be determined by the force necessary to work the pump or by setting the engine a little past the center and holding the flywheel, noting when it begins to pull. When the required pressure is obtained, the flywheel should be turned until the pin is about 10 degrees past the center and the charge fired by means of a match igniter or by a spark of the electric igniter.

In large engines, where the horsepower is 100 H. P. or above, a small gas engine, which may be easily started by hand may be employed to turn the larger engine over

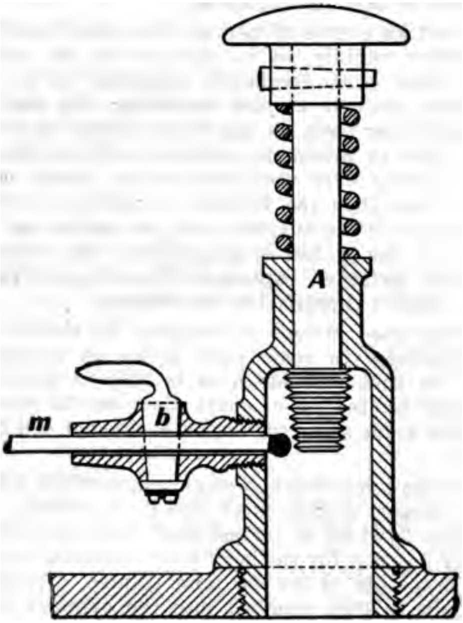

Fig. 35.

until it takes up its cycle, and the starting engine is then uncoupled automatically. Sometimes the valve mechanism of the engine is so arranged that, after the engine is uncoupled from the line shaft and before it is stopped, it is employed to charge a pressure tank by permitting a portion of the expanding charge to pass into the tank. This is accomplished by opening communication with the tank after the piston has made a portion of its expansion stroke, the first portion of the stroke being used to impart motion to the engine.

Large engines are nearly always started by means of compressed air, and in case of an extensive power plant, the use of a gasoline engine for driving the compressor alone is sometimes employed. This engine may run a belted compressor, it may be direct connected, or the compressor and the engine may be built as one unit.

In starting automobile motors, several methods are employed. The most customary one is to throw on the current at the switch and rotate the spark control lever until the timer causes a spark at one of the plugs, or to short-circuit the timer by means of a push button. Another method is to store pressure by means of a special check valve attached to the cylinder, allowing a part of the products of combustion to flow to a storage tank. This gas is admitted to the engine either by a distributing valve, or by means of valves on the cylinders operated magnetically through the timer. Still another method is the use of a spiral spring which is wound by the power of the engine or a special crank, and reducing gears. This method however, has found very little favor.

In the marine engine of moderate powers, a reducing gear is frequently employed; sometimes this consists of a pair of open gears, but most generally a pair of sprockets and a chain is used and the starting crank is elevated to be within convenient reach of the operator. In connection with such devices, there is frequently used some form of safety crank. This is a device to prevent injury to the operator in case the engine is started with the spark too far advanced. It is so arranged that when the motor starts backward, the crank is disengaged and the motor will turn freely. The author strongly recommends a device of this kind on all engines that must be started by hand cranking.

The ninth edition of the Handbook sees the electric starter quite universal in its application to the automobile. It has practically superseded all other types. The electric starter consists of three parts, a generator, a motor and a storage battery.

The majority of the starters in use employ what is known as the two-unit system. In this system the generator or dynamo and the electric motor are two separate pieces of mechanism. The generator is permanently geared to the engine and the motor is arranged with a special sliding gear, which is thrown into mesh with a gear on the engine at the moment of starting, and automatically unmeshed so soon as the engine picks up its cycle. In the majority of cases the gear is attached to or forms a part of the flywheel.

In the second system the generator and the starting motor are combined into one, and the device is permanently geared to the engine. This is known as a single-unit system. The efficiency of this system is generally considered inferior to the two-unit system. In has the advantage of compactness and simplicity.

The battery used is either a six-volt or a twelve-volt combination. Generally speaking, the twelve-volt system is that most extensively used.

The electric starter is now coming into use to a considerable extent for airplane engines. For the average engine the weight of the generator, starting motor and switch approximates 50 lbs. The battery, of special design, weighs about an equal amount. Therefore, an electric starting system adds about 100 lbs. to the weight of the power plant.

CHAPTER XIV.

ENGINES FOR AUTOMOBILES.

The requirements of an automobile engine are lightness, compactness, minimum vibration, wide range of speed under control of the operator, and simplicity. Economy of fuel consumption is also an important consideration. The fewer the cams, levers, etc., on an automobile engine, the less liable it will be to get out of order. It was formerly supposed by the automobile builder that a gas engine must run at a constant speed and all changes of speed made by means of gears. This is very amusing in the light of later experience, but in the early days of the industry it was insisted that controlling the speed of the motor itself was beyond realization. Speed regulation ranging from 150 to the maximum speed of the engines is now common. Two methods are employed for governing the speed of the engine. One is by throttling the charge, and the other by changing the lead of the spark. Both methods are employed in conjunction, but in engines fitted with a governor, the governor usually controls the throttle alone. Some makers attach a governor to the timer. However, this is not for speed control, but to advance or retard the spark automatically according to the speed of the engine. Either of these methods gives an impulse at each cycle.

For single-seated cars, an engine capable of giving at least 16 horsepower at 1,800 r. p. m. should be employed. If the automobile is likely to be called upon for heavy hill climbing, the horsepower should be increased to from 20 to 30. For motor bicycles, 2 to 3 horsepower is usually sufficient.

A very good rule, is to employ one horsepower for each one hundred pounds of car. A great deal, however, depends upon the service for which the car is intended. For the ordinary runabout or town car this rule gives good satisfaction, but for country touring, especially when the car has large carrying capacity, 1½ horsepower per 100 pounds will give better satisfaction.

The greater number of the earlier automobiles were fitted with single cylinder motors, then came cars with two cylinders, to be followed by a few threes, fours and finally sixes, in the pleasure vehicle, while today as many as twelve cylinders are being used. The favorite number of cylinders for the average car is four, on account of the smooth running and freedom from vibration. The six-cylinder, especially in the four-cycle type, gives an extremely even torque, and for large cars is in considerable favor, while eight and twelve are common.

The greater number employ the four-cycle type. As for the two-cycle, it is no longer in favor at the time of writing this (the ninth edition). The two-cycle type has proven difficult to develop along automobile lines, and its progress as an automobile motor has been slow because more attention has been paid to the older type. But, as the author will show, in another chapter, the two-cycle when properly designed and carefully built deserves recognition.

Among the automobile builders, the jump-spark is in greatest favor, as it may be better controlled at the high rotative speeds at which these engines run. The hammer-break had some vogue, but the proportion of cars using it was small. The change of lead for the spark may be obtained by rotating the cam or the timer, as the case may be, around the camshaft. Governors for automobile engines are used on quite a number of the larger cars, and avoid the close watching of grades necessary without them. They are invariably placed under control of the operator that he may vary the speed setting at will, and generally they may be put out of action altogether if desired. The greater number of automobile engines are controlled by the operator without the intervention of a governor.

Cooling of the cylinder walls is accomplished both by water and by air. In a water-cooled motor, the water must be passed through some form of radiator for cooling. For circulating the water, either a pump or the natural circulation of the water from the heat and known as "thermo-siphon" circulation is employed. In air-cooling, the cylinder wall is covered with ribs or pins, and the heat taken up by the air directly from the cylinder wall. In both systems a fan is frequently employed to increase the air circulation, and in some systems of air-cooling, the cylinders are covered with jackets and air drawn through them by means of a blower.

On the smaller cars the gasoline tank is generally placed above the carbureter and the gasoline flows to it by gravity. If the tank is placed low, as on many of the large touring cars, a vacuum system is employed to raise the fuel to the carbureter. Lubrication of every part should be strictly automatic, so that when the engine has been given the proper attention at the outset of a trip, it will need no further attention for a reasonable time at least. The fuel feed should be so regulated that no odor of unburned gasoline is given off at the exhaust. The muffler should be a good one, as excessive noise is an objectionable feature. It is necessary to sacrifice a little power to get good results with the muffler, but with careful design, a muffler can be made that will give little or no back pressure.

CHAPTER XV.

MARINE MOTORS.

As compared to the automobile motor, the marine motor must be designed to stand up to hard work, and practically full speed continually for hours at a time; whereas the automobile motor is very seldom used at full power except when climbing a stiff grade or working through heavy roads, and then but for a few moments, only. The marine motor is almost always run at its highest possible speed. For this reason certain proportions, as for example, the size of the bearings, which are quite ample for an automobile motor, are insufficient in marine service, and for this same reason, an automobile motor is very seldom successful when used for marine purposes.

The various proportions which are usually required for marine service will be dealt with more in detail in the chapters on design, and will be touched on briefly at this time. The designer should note, however, that there are several things about a marine motor and its requirements that do not appear in other motors, and these are, in general, as follows:

Marine gasoline motors are used in three classes of boats and on boats of all sizes, from a tiny canoe or dinghy to ocean-going power craft of large proportions. The waters on which motor boats are used are practically all of those in which there is depth sufficient to float a craft of any kind. There are, for example, two kinds of water, fresh and salt. It must be remembered that for fresh water, iron propellers, steel shafts, etc., may be used without causing trouble, but for salt water such parts as come in contact with the water, as in the cooling circulation, should be of bronze, or they will soon be so corroded as to be put out of service. Then, there are different classes of the several waters, which may be classed as clear and foul waters, such as the many rivers we find so frequently in the West, and shallow bays, or rivers with sandy bottoms which impede the pumps and clog up the water jackets, and there are other foul waters filled with vegetable matter, waste from mills, etc., which should be taken into account and provided against if possible.

In no branch of the motor industry do we find so many two-cycle motors as in motor boating. It is not many years since the two-cycle predominated in this field, and even today, the author believes, that in the smaller marine motors, those below 40 H. P., the two-cycle still predominates. This is due chiefly to the fact that they can be more cheaply constructed than a four-cycle and having greater simplicity show, in the more modern designs of two-cycle, less tendency to get out of order than the four cycle type. While, with the greater majority of the smaller and poorly designed and poorly constructed two-cycle motors, the fuel consumption is much greater per horsepower hour than a four-cycle, fuel is of little importance when the motor is small. However, in the more recent designs of motors of this type, the fuel economy approaches closely if it does not reach, that of the four-cycle motor.

For canoes of the ordinary pleasure type of say, 14 to 16 feet, a single cylinder 2 H. P. two-cycle motor is the size most generally used. In the 16 to 18 ft. racing canoe, we seldom find more than 6 H. P., and as a rule, the larger size of canoe motors has two cylinders. In some of the smaller launches, which are most generally power skiffs or row boats, from 2 to 4 H. P. single cylinder motors are those most generally used. This is also true of the power dinghy, or other form of yacht or sail-boat tender. As we go into the larger launches, we find for pleasure boats, from 3 to 10 H. P. in lengths of from 18 to 20 feet, and in boats of the 25 ft. class, either open or cabin, an average of 10 H. P., while in the 18 to 20 ft. semi-speed boat, 15 to 25 H. P., generally of three to four cylinders, and, in the high speed racers, of the 20 ft. class, which are generally of the hydroplane type, motors ranging all the way from 20 to even as high as 100 H. P. or as much power as can be crowded into the various weights permissible in boats of this size.

Cabin cruisers of the smaller sizes, say from 20 to 25 ft. in length, and usually made of the trunk cabin type, carry generally from 10 to 20 H. P., while larger open and cabin boats from 30 to 35 ft. in length may carry from 10 to even as high as 60 H. P., according to their requirements as to length of journey or speed desired. The larger boats vary greatly in their power requirements according to the conditions to be met.

In the marine motors for ordinary pleasure purposes we very seldom find a single cylinder motor greater than 10 H. P., and as a rule, we find the two cylinder motors beginning at about this H. P. and running as high as 20; three cylinder motors in powers of from 12 H. P. to 30; four cylinder motors of from 16 to 40, and six cylinder motors in powers from 30 to 60 for boats of the smaller sizes. As we go into the larger powers we usually find the motors of from four to six cylinders, eight cylinders being seldom used except for boats of the absolute racing type, in which case, however, motors of from eight to even sixteen cylinders are occasionally built. In the high powered cruisers, the six cylinder motor seems to be the favorite and in sizes of from 40 H. P. up, the four cycle motor predominates.

For pleasure boats we find the smaller motors, usually of the medium to high speed type, running from 400 to 500 r. p. m., to as high as 900 and 1,000, while in boats of the absolute racing type, especially in the smaller powers of 100 H. P. and below, speeds as high as 1,200 or even 1,400 r. p. m. For service boats, also called working and commercial boats, and used for such various purposes as commercial fishing, freight carriers, towing, etc., we generally find comparatively slow speed motors, running at speeds of 500 r. p. m. and below, and even as high as from 20 to 25 H. P. in a single cylinder motor. As in the pleasure boat, we find the smaller service boats using a two-cycle, but in this class of boat, a two-cycle frequently runs into powers of 60 H. P. and more. For stern wheel boats used so much on our narrow rivers in the middle West and South, we often find a heavy single cylinder horizontal stationary engine equipped with a belt drive to a jack-shaft, whence the power is taken to the stern wheel by either one or two chains, and the reversing accomplished by either a straight or a cross belt.

For the smaller boats we find engines coupled directly to the propeller shafts, and arranged to reverse by means of the spark. On motors of 5 H. P. up, however, it is usu*ally the custom* to employ a reverse gear while in the very large sizes, both starting and reversing are accomplished by means of compressed air and special valve operating mechanism. The gasoline motor has become such a large factor in power boats, we now seldom see a steam pleasure craft under 75 feet in length.

As we pass to the larger ocean-going craft, we find the oil engine in extensive use. In the smaller powers the hot-bulb engine is the type most frequently employed. In the very large sizes, the Diesel predominates, while in the submarine, so prominent in the European War, the Diesel is employed to the exclusion of all others.

In the hot-bulb engine the two-cycle is, with one or two exceptions, the only type employed. There are quite a few two-cycle Diesels, but the majority are four-cycles.

CHAPTER XVI.

AERONAUTICAL MOTORS.

The rapid strides in aviation made in the recent years are but the culmination of years of experiment. The principle of the airplane has been known to scientists for a long time. The only great draw-back to the actual accomplishment of mechanical flight has been the lack of a suitable power plant. Maxim, as early as 1894, had a steam power plant which was a wonder for its small weight for the horsepower developed, the weight of the entire power plant of 350 H. P. being less than ten pounds per horsepower.

It is quite possible to get an airplane off the ground and make fairly good flights with a power plant as heavy as this, but the best professional aviator of to-day calls for a power plant of 5 pounds per horsepower or less, exclusive of fuel supply. In the air-cooled motor of the revolving cylinder type this weight has actually been brought down to as light as 2.2 pounds per horsepower for a one hundred horsepower motor. This is particularly noteworthy for the reason that the motor is of the four-cycle type, and therefore each cylinder receives an impulse but once in two revolutions. In the water cooled type of four-cycle airplane motor the weight usually runs very close to five pounds per horsepower and quite frequently exceeds it when the weight of the radiator and necessary cooling water is considered.

A much lower weight per horsepower can be secured in a motor of the two-cycle type. For example, in a 50 H. P. two-cycle motor designed by the author, the actual weight of the motor itself is 173 pounds; radiator, 34 pounds; water, 20 pounds, a total weight of 227 pounds, a little over 4½ pounds per horsepower.

Usually, as the horsepower increases, the weight per horsepower decreases slightly, for example in a similar design of 75 H. P., the weight for the motor alone is 243 *pounds, or 3¼ pounds* per horsepower.

Few designers appear to realize that the service required of an airplane motor is most severe, comparable only to that required of motors used for racing, both in automobiles and motor boats. In an airplane motor, more than any other type, must the possibility of a break-down be guarded against, for the stoppage of a motor in the air may necessitate a landing on rough ground, or even in forests, or among buildings or possibly in the water.

Furthermore, the airplane motor is run at an extremely high speed, and frequently at an angle from the horizontal that is seldom, if ever, found even in an automobile motor. As the confidence of the aviator increases, he makes flights of long duration, often for hours at a time, and with the motor running practically to its limit of capacity. To produce a motor that will meet these conditions satisfactorily, and at the same time be within the required limits of weight, is a most difficult problem in design, and calls for the greatest skill of the constructor.

The greatest mistake that is made by the average designer of an airplane motor is to sacrifice the strength and operating qualities of the motor to bring it within the required weight limit. As we profit greatly by our mistakes, I shall point out briefly the errors generally made by the average designer in getting out a motor of this class. One of the most frequent is to give the motor insufficient cooling surface, and we find quite a number of motors troubled with over-heating simply because the water jacket is not long enough, and the circulating pump is of insufficient capacity, so that no matter how large the radiator, the motor cannot be run for any great length of time without over-heating. In the four-cycle motor, the water jackets should be carried at least as far as the lower end of the piston travel, measuring, of course, from the top of the piston. In a two-cycle motor the jackets should be extended at least twenty per cent below the top of the piston when the latter is at the bottom of its stroke. The radiator, when of the ordinary honey-comb type, should have at least 2½ square feet of cooling surface per horsepower, but in order to make this effective the pump should be of sufficient capacity to ensure good circulation.

A certain well-known make of two-cycle airplane motor was notorious for the fact that a flight of over ten *minutes* was unusual without the motor over-heating. Another mistake made in designing this class of motor is making the crank-shaft too weak, and the flanges on both the cylinders and the base of the motor insufficient to stand the strain. Another mistake is the use of short bearings on the crank-shaft and the crank-pin which will not remain cool, even when flooded with oil. The size of an airplane motor crank-shaft will be discussed fully in the chapters on design. Still another mistake is the placing of the thrust bearing for taking the thrust of the propeller between the web of the crank-shaft and the crank-shaft bearing farthest from the driving end, in other words, between the web of the shaft and the forward bearing so that the thrust is taken through the *entire* length of the crank-shaft. The bearing at the propeller end is frequently made too short when it really should be fifty per cent longer than the common practice.

For practical aviation, the airplane of the two-propeller type, like the Wright, requires the smallest power. In fact, greater efficiency in the propeller is obtained by running it more slowly than the motor. Powers as little as 25 H. P. have flown airplanes successfully, but for the average type of monoplane and biplane, from 40 to 50 H. P. is required and for meeting all sorts of weather conditions, getting off the ground where it is rough, and in a short space, the author recommends from 60 to 80 H. P. With an ordinary single or two passenger machine and an engine of this power, the speed of the engine may be reduced after getting off the ground and, since the engine is not being driven to its full capacity, it will stand up better. As this edition of the Handbook goes to press, aviation is calling for larger powers, and motors of from 125 to 350 H. P. are in demand for fast scouts and battle planes.

Now that aviation has so far advanced that it is not so much a question of whether a machine *will* fly as *how* it flies, the efficiency of the airplane is receiving attention. Aeronautical engineers are giving special attention to head resistance, and its reduction to a minimum. The refinement of balance and of control is making rapid progress and the efficiency of lifting surfaces is being increased.

The airplane is passing through the same stages of *development as did* the automobile in the early days of the century, although the progress would appear to be more rapid. These refinements have already shown that the horsepower required to fly is much less than was originally thought necessary. It would appear that for an equal number of passengers the power required in the future airplane will be about the same as it is customary to employ for a high-class automobile. That we are rapidly approaching such a condition in aeronautics is shown by recent developments.

The appearance of the ninth edition of the Handbook finds us in the fourth year of the great European War. In the past three years the advance of aviation has exceeded the advance made in all time previous to the war. The impetus the conflict has given to this science has placed the machine on a par for safety with the high-grade motor car. The improvement in airplane engines has been gratifying, and they are most dependable, but not durable. Much improvement is desirable along the line of durability.

CHAPTER XVII.

LUBRICATION.

The extremely high temperature within the cylinder of a gas engine, makes the use of special oils for its lubrication a necessity. The oils employed for the cylinders of steam engines are not suitable for the gas engine as they will gum and coat with carbon the piston and the interior of the cylinder in a gas engine. Many of the vegetable oils and practically all of the animal oils are unsuited for this purpose on this account. For this reason petroleum oils of a high fire test and free from carbon are employed. Oil having a high fire test is one that must be heated to a high temperature before it will take fire at the surface. This point should not be confused with the flash point, which is the temperature at which the vapor given off by the heated oil will take fire.

The cylinder oil should flow readily at low temperatures for winter use and yet it should not vaporize at the high temperature of the explosion, as this would destroy its lubricating properties. Oil for summer use is generally heavier than for winter use, although the author prefers an oil of medium heavy body for any season of the year. Oil for air-cooled motors is usually of a higher fire test than for water-cooled motors, and I have found that for high speed motors, such as those of the automobile type and especially for high speed motors of bores above four inches, a better grade of oil is necessary than for small motors. Oil that will gum or carbonize rapidly clogs up the passages, makes the rings stick in the grooves and frequently will deposit in such a manner as to cause an incandescent point and produce premature ignition. Any deposit of this character will cause the motor to be troublesome. If the rings stick, the compression will be reduced. If the passages are clogged, (the exhaust) the gases cannot get out. If prematures occur the motor is *likely to backfire* through the intake passages, or at least have back pressure. All of these tend to reduce the power of the motor as well as cause it to work erratically.

In air-cooled motors of the airplane type, lubrication is generally effected by the use of castor oil, not the so-called mineral castor oil, but the product of the castor bean. This is used in large quantities, usually one-third as much oil being employed as gasoline. The function of the oil in these motors is in but a small proportion a lubricant, and in a greater proportion cooling. To attempt to operate them with ordinary oils employed for the average gasoline motor will cause them to over-heat in a very short time. The castor oil passes through the motor so rapidly that it has not time to carbonize.

The fact that an air-cooled airplane motor will heat rapidly with the ordinary lubricants while an air-cooled automobile or cycle motor may be operated with them successfully, appears inconsistent. It should be considered that the two operate under different conditions, the aeroplane motor is at practically full load and full charge all the time it is running while the automobile motor is operated at its full load for short intervals of time only. Castor oil is required for air-cooled airplane motors only. The water-cooled type will give good service on high quality petroleum gas engine oils and the proportion of oil to gasoline is from one thirty-second to one fortieth.

Graphite has proven itself useful in many forms of lubrication and for the bearings of the shaft it gives good service in the ordinary form if not allowed to creep into the cylinders where jump spark ignition is used, as it would foul the plugs. There is a form of graphite that will mix with and stay in suspension in oil and not foul the plugs if used in moderation. By its use the amount of oil fed to the cylinders may be cut down one-half to one-third. It will not, however, stay in suspension in oil nor in oil mixed with gasoline when either are acid.

Generally a gas engine cylinder oil gives fairly good results as a machine oil and may be employed for the bearings as well. For a large engine where the oil required for the bearings is considerable, a cheaper machine oil may be employed to advantage. For small engines, using the *same* oil is a great convenience, and for splash *or forced* and flooded lubrication it is necessary to use one *oil.*

Cup grease is a very good lubricant for bearings. With automatic cups for large bearings and the plain compression grease cup for small bearings, it is cleanly, economical and automatic. It is not thrown out from the ends of the bearings like oil. A little of it will last a long time and it is not expensive in first cost. It is automatic because, if the bearing heats up, the fluidity of the grease is increased and it will feed to the bearing faster. For marine engines especially, it is desirable on this account, and particularly where the boat is small and the seating space surrounds the engine.

Cup greases are made from animal fat, from the petroleum vaselines and from the two in combination. They are frequently charged with other substances for various purposes such as graphite for greater lubricating efficiency, cork dust for gear cases to cushion the noise of the gear teeth, etc.

The application of the lubricant to the various parts of the engine is often considerable of a problem for the designer. This is especially true of the high speed motor whether it be of the automobile or the marine type. The parts most difficult to reach are the crank-pin and the connecting rod bearing on the piston pin, in a motor with a closed base. For the four-cycle motor splash lubrication gives fairly satisfactory results. In this form of lubrication the base is filled with oil to a point that will just allow the lower end of the connecting rod to strike the top of the oil. This splashes the oil about the interior of the base over the wall of the cylinder and on the crank-pin and the piston-pin bearings. For the two-cycle motor with base compression, splash lubrication does not give satisfaction because the oil is carried up the transfer port and out through the exhaust passages, and the bearings would soon be dry were not the oil fed continuously.

The modern manufacturer of high speed motors, especially those of the four-cycle type, is depending more on forced-flooded lubrication. In this system, the base is so made that it slants to one collecting point, an oil sump from which the oil is picked up by a pump and forced directly through the various bearings keeping them flooded at all times with an excess of oil. For the cylinder and piston this form of lubrication in excess, would cause the motor to smoke and for this part the oil is usually pumped into a groove into which the piston dips at the bottom of its stroke.

For the two-cycle motor with enclosed crank-case in which the charge is compressed before it passes to the cylinder, the lubrication of the crank-pin has long been a serious problem. The use of a wick projecting from the lower bearing, together with a constant feed of oil to the base gives good results. Backfiring in the base, however, or an over-heated bearing will char the wick, so that it will not carry the oil to the bearing. If not renewed, the wick is then a detriment rather than a help. Another plan of oiling the crank-pin is to drill a hole through the center of the pin in a slanting direction, through the arm and thence out through the main bearing. Through this hole the oil from the main bearing is forced to the pin. Another method is use of an oil ring which picks up the overflow from the main bearing and by centrifugal force sends it to the pin bearing through a hole in the end of the pin. These various plans are illustrated in Fig. 36.

For the two-cycle motor, all the methods above described have their objectionable features. But there is fortunately an exceedingly simple method of oiling the interior of a two-cycle motor that is at the same time absolutely dependable. This method is to mix the lubricating oil with the gasoline and allow it to enter the engine through the carbureter. The oil will dissolve in the gasoline just as sugar will dissolve in water, and it remains in solution. When the oil enters the base of the motor it deposits on the walls of the case and the inside of the cylinder. The lower end of the connecting rod should be merely a strap so that the pin will be bare and the oil will then reach it effectively. It should be remembered that the rod in a two-cycle motor is in compression always, and therefore there is no bearing pressure on the lower half of the rod. This method may be carried to the extent of lubricating the main bearings by providing proper oil channels. The author believes that, especially where the main bearings are long, grease cups should even then be used as a safeguard. The oil will, however, be forced into the bearings by the base compression and the grease cups on the shorter bearings between the bases appear to feed but a very little. This method of lubrication is not only effective, but absolutely dependable. The proportion of

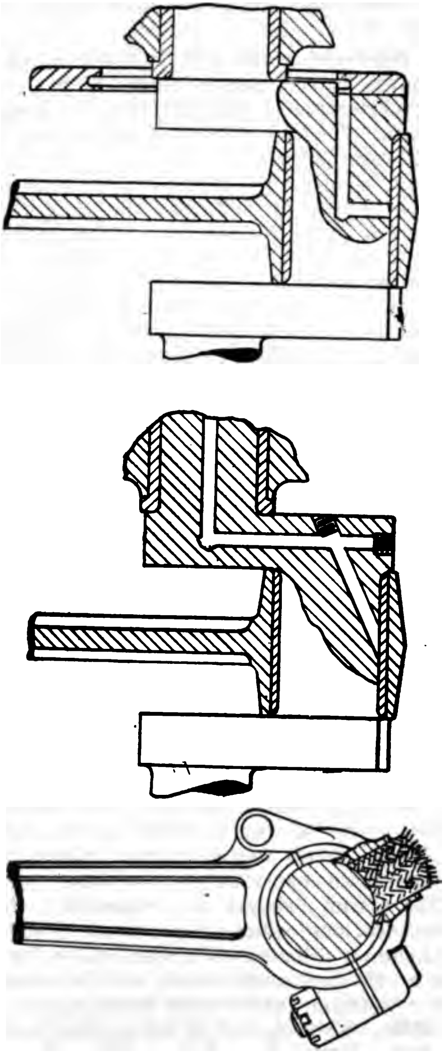

FIG. 36.

oil to gasoline varies from one pint to four gallons for a new engine to one pint to five gallons on an engine that has been well run in. In two-cycle airplane motors the author recommends excess of oil, and as high as one quart per gallon has been used. Two quarts to each five gallons appears to give excellent results. The method has been found useful in four-cycle motors as a safeguard, to insure oil reaching the cylinders.

PART II.

DESIGN.

CHAPTER XVIII.

THE INDICATOR DIAGRAM.

With the aid of the indicator diagram secured by means of a modified steam engine indicator on engines of moderate speeds and by means of the manograph on high speed engines, we can determine accurately the pressure at any point of the stroke. With this information we can not only analyze the performance of the gases within the cylinder, but at the same time learn much of value that will assist in overcoming difficulties that are otherwise a mystery to the engineer and the designer.

If the designer could have before him an ideal diagram and knew the conditions necessary to produce such a diagram, he would have an excellent foundation upon which to base his design. The author by no means wishes to be understood as believing that the securing of an ideal diagram should be the sole aim of the designer. It is not to be denied, however, that a thorough study of this important index of performance is shamefully neglected by a great many. I shall therefore show in this chapter, not only the ideal diagram, but how the indicator diagram will point out many faults in design.

An ideal diagram for the average gas engine is shown in Fig. 37. In laying out a diagram it is customary to select the compression found best by experience for the fuel and the conditions under which it is used. For an average condition, the water-cooled gasoline engine of medium speed has been chosen in which the compression for best results has been found to be about 85 pounds per square inch. The designer knows that to secure this compression the ratio of the volume of the compression space should be approximately .3 or 30 per cent. For the purpose of this illustration it is more convenient to start the diagram at this end of the problem and therefore for the

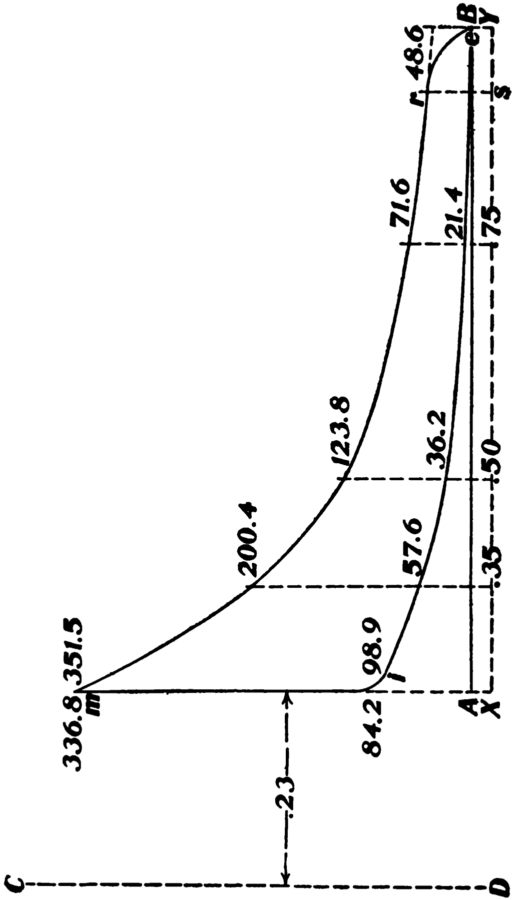
C
D
.23
336.8
m
351.5
84.2
98.9
l
200.4
57.6
123.8
36.2
71.6
21.4
r
48.6
B
e
Y
s
.75
.50
.35
A
X

FIG. 37.

diagram in Fig. 37 the ratio of the volume of the compression space to the volume displaced by the piston has been selected as .3.

The total volume of the cylinder when the piston is at the end of its outward stroke is therefore $1 + .3 = 1.3$ times the piston displacement. As the calculations may be made more readily by considering the total cylinder volume as unity, the first operation is to find the proportion of the total cylinder volume that is included in the compression space. Dividing .3 by 1.3 gives .2308 as this ratio.

For the purpose of calculating the pressures at various points of the curves the author has evolved the following formulas from the average curves of a number of indicator diagrams. For the compression curve.

$PV^{1.3}=K$ (1)

Wherein P = the pressure above a vacuum, absolute pressure;

V=the volume of the gases at the time they are at the pressure P;

K= a number or constant depending upon the conditions, but which is the same for all parts of the curve.

When the volume of the cylinder is taken as unity, K becomes the absolute pressure of the atmosphere, usually taken as 14.7 pounds per square inch. For the maximum pressure after explosion, an examination of quite a number of indicator diagrams shows this pressure to be about four times the compression pressure. The ratio appears, however, to vary considerably with both the nature of the fuel and the compression pressure. For our present purpose this ratio will be taken, based upon the pressure above the atmosphere.

The formula for the expansion curve evolved in the same manner as formula (1) is

$PV^{1.35}=C$ (2)

Wherein P and V have the same significance as in formula (1) and C is a constant depending on the maximum pressure.

These equations may also be written as follows:

$$PV^{1.3}=P_1V_1^{1.3}$$
$$PV^{1.35}=P_1V_1^{1.35}$$

a form which shows the exact relation between any two *points on the curve.*

That the reader may better understand the application of these formulas the computation of the diagram in Fig. 37 will be given. Taking first the computation of the pressure at the end of the compression stroke, and applying formula (1)

$PV^{1.3}=K=14.7$, and

$$P=\frac{14.7}{V^{1.3}}=\frac{14.7}{(.2308)^{1.3}}$$

In order to find the denominator of this fraction it is necessary to multiply the true logarithm of the value of V by 1.3. The logarithm of .2308 as found in the table of logarithms (the tabular logarithm) is $\bar{1}.363236$. Since this logarithm has a negative characteristic (number to left of decimal point), the true logarithm must be found by adding the mantissa (number to right of decimal point) to the characteristic as follows:

$$\begin{array}{r}
-1.000000 \\
.363236 \\
\hline
.636764 \\
1.3 \\
\hline
1910292 \\
636764 \\
\hline
.8277932
\end{array}\ \text{true log of } (.2308)^{1.3}$$

In order to get the tabular log subtract the true log from 1 and add —1, thus:

$$\begin{array}{r}
1.000000 \\
.827793 \\
\hline
.172207 \\
-1. \\
\hline
\bar{1}.172207
\end{array}\ \text{tabular log of } (.2308)^{1.3}$$

Subtracting this log from 1.167317 the log of 14.7 we have

$$\begin{array}{r}
1.167317 \\
\bar{1}.172207 \\
\hline
1.995110
\end{array}=\log 98.888=\log P.$$

As P is in pounds per sq. in. above a vacuum, it is necessary to subtract 14.7 to get the gauge pressure. Taking P as 98.9, the gauge pressure will be 98.9—14.7=84.2. To secure the probable maximum pressure, multiply by 4 *getting the* gauge pressure at *m*, Fig. 37 as 336.8 lb. and *the absolute* pressure as 336.8+14.7=351.5.

In order that the proper shape of the compression curve may be closely approximated, it is best to compute the pressures at various points on the curve. In the figure these points are taken, .35, .50 and .75 of the total volume. The pressures corresponding to these points are 57.6, 36.2 and 21.4 lb. respectively. In case of any uncertainty as to the shape of the curve it may be well to calculate the pressure at points between, and especially at a point midway between .35 and the end of the curve.

The pressures for the expansion curve are found in the same manner as for the compression curve, but by means of formula (2). Before this formula is applied, however, it is necessary to find the value of C, which is the pressure of the expansion line at the end of the stroke when the volume is equal to 1. Hence $PV^{1.35}=C$ becomes $351.5\times(.2308)^{1.35}=C$.

By the same process as for the compression curve formula, log $(.2308)^{1.35}$ is found to be $\bar{1}.140368$. Adding this log to 2.545925, the log of 351.5, the sum is 1.686293 and from the table this is found to be the log of 48.56. Hence $C=48.56$, which is the terminal pressure of the gases at the *end* of the stroke provided release does not take place before that time. This gives the equation of the compression curve as

$$PV^{1.35}=48.56.$$

The intermediate pressures for the remainder of the expansion curve are found by subtracting the logs of .35, .50 and .75 raised to the 1.35 power from the log of 48.56 and finding the corresponding numbers in the log table. These pressures are 200.4, 123.8 and 71.6 respectively. After these points have been located, the diagram is constructed by drawing curves through them as shown in the figure. The length of the diagram shown in the figure is 3½ inches from the line mX to e and the scale of the spring represented is 160 lb. Hence unit volume is represented by a length of 3½ inches and each inch in height represents a pressure of 160 lb. per sq. in.

In order to make the diagram complete, it is necessary to add the two small curves at i and at re. The curve at i represents the sudden increase in the rate of pressure increase due to firing the charge before the end of the stroke. It is more pronounced the greater the lead of the *ignition, i. e., the* distance before the end of the stroke that the charge is ignited. For this reason it can be shown but approximately. The effect of this curve upon the area of the diagram is too small to be of consequence. Its principal value is to serve as memorandum when comparing this diagram with those actually taken from an engine.

The curve *re* represents the sudden increase in the rate of the fall of pressure due to opening the exhaust valve before the end of the expansion stroke. This early release of the exhaust is necessary to avoid back pressure during the return stroke during which the exhaust is being expelled by the motion of the piston. The point of release *r* should be so chosen as to bring the point *e* where the expansion line meets the atmospheric line at the end of the stroke. If it occurs too soon, power is dissipated by the exhaust, and if too late, back pressure results.

As with the ignition point *i*, the point of release will vary according to the conditions of pressure, speed, etc. For engines of moderate speed it will usually be found at one-tenth of the length of the diagram from the end. To draw this curve as shown in the figure, erect the vertical line *sr* at a distance from *Xm* equal to .9 of the length of the diagram, which in the figure is $.9 \times .77 = .693$ times the unit length. Through the points *r* and *e* describe a circular arc tangent to the expansion line at *r*. Then join the curve *i* and point *m* with a straight line to complete the diagram.

After completion of the diagram, the designer may find the M. E. P. by means of a planimeter as explained in the chapter on testing. From the M. E. P. obtained from the diagram the designer may calculate the horsepower of an engine it is proposed to build or he may determine the bore and stroke of the engine that will give a certain required horsepower at a given speed. The design of the valve motions may, to a certain extent, be founded on the diagram, as it gives an indication of the proper time for opening and closing the valves. It should be the aim of the designer so to construct a gas engine that it will produce a diagram approaching very closely that shown in the Fig. 37, in its general form. It must be remembered, however, that the ideal diagram for each fuel will vary from that shown in the figure both in compression and in maximum pressure after explosion. Know-ing the particular governing conditions for any fuel or special cycle, the designer may, by following the model shown, lay out an ideal diagram to fit these conditions for his guidance.

In order to illustrate how these conditions vary, a few examples from actual diagrams will be given. The pressure ratio is the ratio of the maximum pressure in pounds gauge to the pressure after compression.

Natural gas, stationary engine; compression 80 lb., maximum pressure 320 lb., pressure ratio 4. H. P. 30, r. p. m. 200.

Natural gas, stationary engine; compression 96 lb., maximum pressure 330 lb. pressure ratio 3.45. H. P. 150, r. p. m. 200.

Illuminating gas, stationary engine; compression 90 lb., maximum pressure 270, pressure ratio 3. H. P. 9, r. p. m. 182.

Gasoline, stationary engine; compression 80 lb., maximum pressure 357, pressure ratio 5.1. This is a special case having no vacuum carbureter, but vapor prepared in gaseous form before entering the suction line.

Kerosene, stationary engine; compression 60 lb., maximum pressure 275, pressure ratio 4.25. Vapor prepared as in the previous case.

Blast furnace gas; compression 127, maximum pressure 242, pressure ratio 1.9. H. P. 700.

Producer gas; compression pressure 83, maximum pressure 248, pressure ratio 3. 90 H. P., r. p. m. 200. Gas high in hydrogen, hence low compression.

Producer gas, average modern practice; compression pressure 100 to 115, maximum pressure 340 to 360, pressure ratio about 3.4.

These figures are from full load diagrams in each case and are taken from various works on the subject. The bearing of these pressures on design will be fully discussed in Chapter XX.

In the two following figures, 38 and 39, are shown diagrams taken from various motors under various conditions, in order to illustrate how to tell from the diagram what is taking place in the engine, and locate the source of derangements or faults in design that cause them.

Fig. 38 gives the actual diagram as taken by means of the indicator with an ordinary indicator spring and shows what takes place during the compression and the exhaust strokes. In Fig. 39 are given diagrams taken with weak indicator springs of about 20 lb. to the inch and on account of their broad intervals show more clearly what is going on during the suction and the exhaust stroke.

Before discussing the nature of the diagrams shown in Fig. 38, the principle causes of variation in the indicator diagram will be given. The time elapsing between the moment when the particles of mixture nearest the igniter are fired until the entire contents of the compression space are ignited is the *period of inflammation.* The speed at which the flame travels to the farthest point of the body of the gas is the *rate of flame propagation.* The rate of flame propagation is greater and the period of inflammation is shorter, the nearer the proportion of combustible and air approaches that for perfect combustion and the higher the compression. Lowering the compression, increasing the proportion of air or burned gases from a previous charge or excess of combustible decreases the rate of flame propagation and lengthens the period of inflammation. The quality of the combustible itself has an effect upon the period of inflammation, and in general the greater the heat value of the gas or vapor, the shorter this period.

Since, for the same conditions of mixture and compression the period of inflammation is the same, the higher the speed of the engine, the less is the time for the completion of one stroke of the piston and the greater is the proportion of the stroke during which inflammation is taking place. Measuring this period, not as a proportion of the piston travel, but as a proportion of the revolution of the crank-pin, it is taken in degrees of arc. For example, for 600 r. p. m. or 10 rev. per second a period of inflammation of 1-240 second would require 15° of crank-pin travel. At 1,200 r.p. m. or 20 r. p. sec. the same period of inflammation would require 30° of crank-pin travel. Hence, to secure complete inflammation at the end of the compression stroke, the time of ignition should be 15° in advance of the end of the stroke for 600 r. p. m. and 30° in advance of the end of the stroke for 1,200 r. p. m.

The reader will readily see that for any change in compression, proportions of the mixture or other conditions that lengthen the period of inflammation, such as lower *compression*, excess of air, etc., the angle of ad*vance should* be increased in order to secure full inflam-

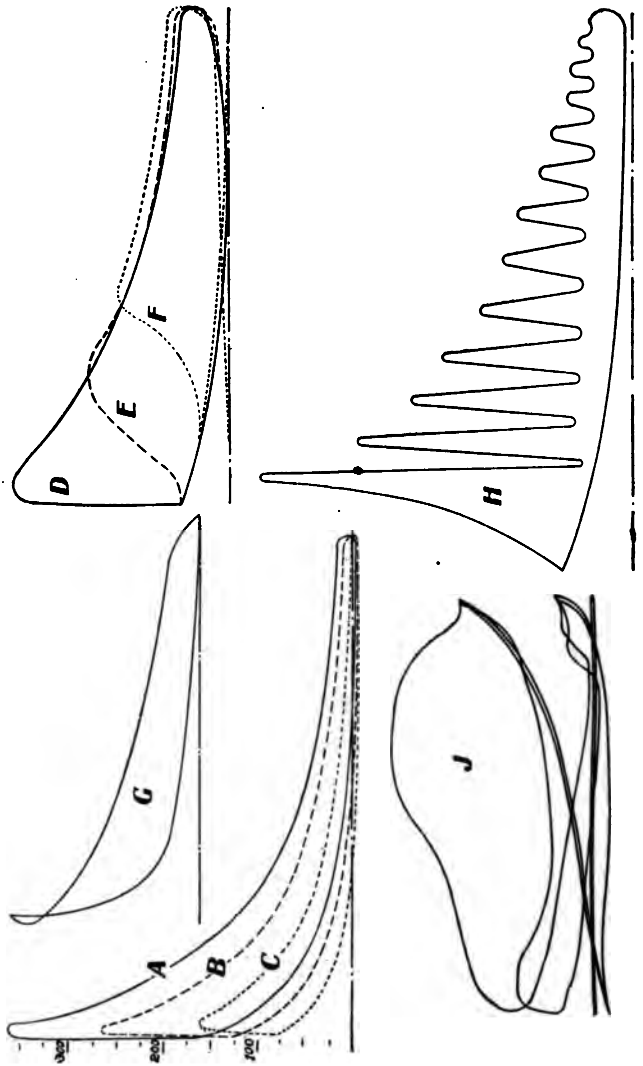

FIG. 38.

mation at the end of the compression stroke. On the other hand, to meet any opposite conditions that shorten the period of inflammation, the angle of advance should be correspondingly reduced to avoid *full* inflammation occurring *before* the end of the compression stroke with the resulting back pressure and loss of power. This change in the period of inflammation and the necessity of altering the angle of advance to meet it, is something frequently overlooked by designers, and the author wishes to call especial attention to its importance in the matter of speed control without loss of efficiency.

Referring now to Fig. 38, diagrams *A, B* and *C* show the effect of throttling the incoming charge without altering the proportions of the mixture or changing the angle of igniter advance. Throttling reduces the quantity of mixture entering the cylinder and consequently reduces the compression, lengthening the period of inflammation. Diagram *A*, shows the throttle slightly closed and *B* and *C* gradually increased throttling. The scale at the left of the diagram shows the compression. It will be noted that the point of maximum pressure is further to the right as the throttling is increased. It is more marked as the throttle nears the position where it is entirely closed. Increasing the lead of the ignition as the throttle closes will straighten up the explosion line (*im*, Fig. 37) and increase the area of the diagram.

The diagrams just referred to are taken from a stationary engine operating at a constant speed. If, however, a variable speed engine such as is used for marine or automobile purposes is controlled by closing the throttle, the effect of the leaning explosion line is not so marked. The reason for this is that as the speed falls off, the ignition is practically correct for the combination of lower speed and lowered compression due to the smaller volume of the charge. Therefore, indicator diagrams taken from this class of motor do not show the importance of ignition advance with throttling and have led to a misconception of this point.

Diagrams *D*, *E* and *F* show the variation caused by late ignition. It should be noted that, while in diagrams *A*, *B* and *C* the compression is lowered as the throttle is closed, in *D, E* and *F* the compression is in each case the same. Diagram *D* shows the ignition very nearly at the correct point for best effect, a slight bending of the explosion line showing near the top. *E* shows the ignition taking place at end of the compression stroke, *i. e.*, on the center, while *F* shows it retarded so that ignition takes place after the center is passed. The effect upon the location of the point of maximum pressure is more marked than in throttling. In the two latter diagrams the period of inflammation is much exaggerated because the speed of the piston is much greater near the center of the stroke than at the ends.

Diagram *G* shows premature ignition, the maximum pressure taking place before the end of the compression stroke. This is a condition that should be carefully avoided as it causes a sudden check to the piston and a knock that is very noticeable.

The exact cause of the wavy line appearing in diagram H has been the source of much discussion among engineers. When it was first brought to the notice of the author, I attributed it to a series of explosions, possibly due to a stratified charge. This opinion was ridiculed so strenuously by a brother engineer that in the first edition of the Handbook, I gave his view of the matter. This was that an extremely sharp explosion set the indicator spring in vibration and produced the wavy line. Subsequently my attention was called to the fact that such diagrams were usually obtained from engines having quite a number of pockets leading into the compression space. My own experience in tests has verified this. One engine in particular, in which the passage from the spark plug to the cylinder was long and tortuous, persisted in giving diagrams of this character. Recent writers insist it is indicator spring vibration.

To show how a diagram will point out the faults of an engine in a marked manner, there is shown in diagram *J*, a reproduction of one taken with a manograph from an experimental engine. This diagram shows two complete cycles or four complete revolutions of the motor. One, the larger, shows ignition about on the center, and the smaller shows an extremely late ignition. That the exhaust valve is opening too late is indicated by the fact that there is no sudden change in the curve of the expansion line until nearly the end of the stroke. That the exhaust valve closes too early is shown by the rise of pressure at the end of the exhaust line. Too small intake passages are indicated by dropping of the suction line below the atmosphere. Too early closing of the intake valve is shown by the sudden drop of the suction line near the end of the suction stroke. The builder of the engine wrote me that he was able as the result of this showing to make a measured improvement in the performance of the engine. The reader will probably agree that the diagram shows decided room for improvement.

As the springs used for the general diagrams have too large a scale to show the suction and exhaust lines clearly, weaker springs with a scale of about twenty pounds are employed for this purpose. These are fitted with a stop to limit the motion of the spring, to avoid damage from the high pressure of the explosion. Such diagrams are shown in Fig. 39.

Diagram *A*, Fig 39, shows the lower portion taken from an engine working at full load. *XY* is the atmospheric line. The line *d-e* is the tail end of the expansion line and shows a drop to atmosphere very nearly at the end of the stroke. There is a very slight back pressure as the end of the line bends toward the left, but this is exaggerated because of the low scale of the spring. The exhaust from *e* to *f* follows atmospheric pressure very closely. From *f* to *a* is the suction line dropping very little below the atmosphere in pressure. The lower portion of the compression line is shown from *a* to *b*. This indicator diagram shows that the engine is operating under practically ideal conditions.

Diagram *B* shows a slight vacuum at the beginning of the exhaust stroke, due, probably, to a long exhaust pipe. The inertia of the gases in this pipe cause a suction in the cylinder which is frequently so strong as to open an automatic inlet valve. The subsequent rise of the exhaust line above the atmospheric, indicates that the passages are small.

Diagram *C* shows too late opening of the exhaust valve and a suction line too far below that of the atmosphere. It will be noted that the exhaust does not reach atmospheric pressure until the end of the exhaust stroke. The excess of vacuum during suction would indicate restricted inlet passages. The reader should consider the conditions under which the engine operated,

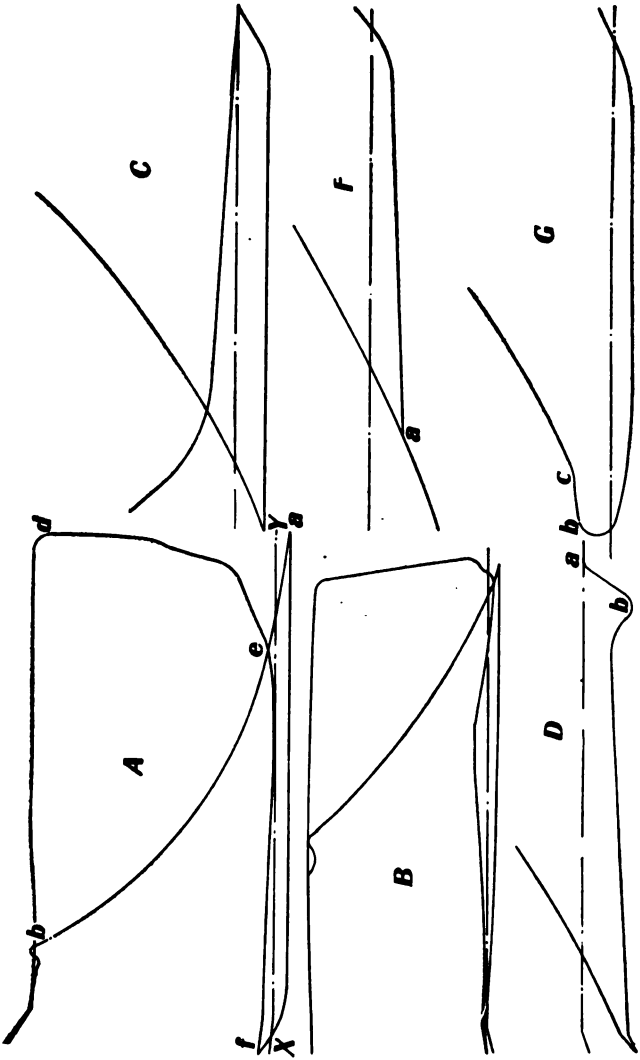

FIG. 39.

when interpreting the suction line. In a gasoline engine, a certain amount of vacuum is necessary in the intake to draw in the gasoline at the carbureter. A suction line parallel to the atmospheric line as shown in this diagram is an indication of fairly good conditions provided it does not fall too low in pressure.

Diagram *D* shows too late an opening of the inlet valve at point *b*. The line from *a* to *b* shows a rapid fall of pressure, too great for normal conditions. The loop at *b* and the rise of pressure indicates that valve has opened and that the suction is steady. But the suction line is not parallel to the atmospheric line, showing restricted passages. In both diagrams *C* and *D*, the inlet valve is closing too soon as the suction line should reach atmosphere at the end of the stroke, bending upward again.

Diagram *F* shows a very early closure of the suction valve, indicated by the rapid fall of pressure from the point *a*; while diagram *G* shows too late a closure of the inlet valve, the line from *b* to *c* indicating that the charge is being forced out through the inlet passages.

In concluding this chapter it should be pointed that intelligent interpretation of the indicator diagram taken from a motor is of the greatest value to the designer.

CHAPTER XIX.

GENERAL DIMENSIONS.

The power of a gas engine is dependent on four main factors; the average or mean effective pressure, the area of the piston upon which this pressure is exerted, the number of times per minute this pressure is exerted upon the piston and the length of the piston stroke. Power computed from the above factors is known as the indicated horsepower (I. H. P.). As there is a certain amount of resistance to overcome in driving the mechanism of the engine itself, the useful power actually delivered to the belt is less than the I. H. P. by the power required to overcome the engine friction and known as the friction load.

The power delivered to the belt is variously termed *developed* or *delivered horsepower.* (D. H. P.) *brake horsepower* (B. H. P.) and *effective horsepower* (E. H. P.). The latter term, on account of the liability to confusion with the term electrical horsepower, is very little used. The term brake horsepower has its origin in the very simple form of absorption dynamometer, the prony brake, used so frequently for determining the horsepower available. Modern practice does not depend altogether on a brake for testing the horsepower of an engine, and as the term *brake* horsepower relates to the method of determining the power of the engine rather than the quality of the power, the term delivered or developed horsepower is generally preferred by engineers.

The ratio of the D. H. P. to the I. H. P. is known as the *mechanical efficiency* of the engine, usually abbreviated to M. E. The formula for this expression is

$$\text{M. E.} = \frac{\text{D. H. P.}}{\text{I. H. P.}}\ .$$

and the result is usually expressed in per cent. Thus, if the I. H. P. of a certain engine is 10 H. P., and the D. H. P. is 8 H. P., the M. E. $= \frac{8}{10} = 80\%$, and the friction *load = 10—8 = 2 H. P.*

What the designer should strive for is to secure not only the greatest amount of power out of a pound of fuel, but also the greatest amount of power for a given bore and stroke. Which of these two points is of the greatest importance depends upon the use for which the engine is intended. In the greater number of cases, and practically always in stationary work, fuel economy is the primary consideration. But motors for racing boats, racing automobiles and aeroplanes have power for cylinder capacity, the primary consideration. Fuel economy is also of a certain amount of importance in these motors, but it becomes secondary to power. Fortunately greatest power for cylinder capacity and fuel economy in engines of both the four-cycle and the two-cycle type frequently go hand in hand, and in striving for one we may attain the other.

Take first the securing of the greatest power for a certain bore and stroke. To secure the greatest D. H. P., the greatest possible I. H. P. and the very smallest possible friction load must be striven for. For a definite speed, the greatest I. H. P. must have the greatest mean effective pressure (M. E. P.). The authentic records of M. E. P.s from the various engines range from 23.1 on the old Hugon engine to 115.4 given by Lucke as obtained from an engine operating on gasoline with a vapor generator instead of a carbureter. The mechanical efficiency ranges in recent designs from as low as 60% to as high as 92%. The author has tested motors which showed as low as 60% on full load and others as high as 84.5%. The average is about 80%, and as a rule this M. E. may be safely taken as the basis of design.

For convenience in estimating the probable D. H. P. to be expected from a single-cylinder motor of a given bore and stroke, the formula

$$D.\,H.\,P. = \frac{D^2LN}{X} \text{ may be employed,}$$

in which D = diameter of cylinder in inches.
L = stroke in inches,
N = r. p. m.,
X = a factor varying with the fuel and the cycle.

The power factor X for four-cycle motors operating *on natural* gas will, for the average size motor, approximate *16,000,* and on gasoline 14,000. As the size of the cylinder increases the power factor becomes smaller, especially where the design of the engine and the nature of the fuel will permit of high compression.

The value of X for four-cycle engines has shown as low a figure as 11,500 on gasoline in exceptional cases. The particular case in mind was an automobile engine with exceptionally large valves and fitted with ball bearings throughout.

For small two-cycle engines operating on gas the value of X is about 12,000 and on gasoline 10,000. Large blowing engines show a value of X as low as 8,400.

Based on the S. A. E. formula for four-cycle engines, H. P. $= \frac{D^2N}{2\frac{1}{2}}$ in which N = the number of cylinders, the value of X is 15,000.

These figures may be employed for estimating the probable power of any engine when the bore, stroke and speed are known. It must be remembered that this formula gives an approximation only and the most accurate way is to determine, from the performance of known engines, the probable M. E. P. and friction load and base the calculations on these figures.

To find the diameter of the cylinder for any engine to give a certain horsepower, the ratio of the stroke to the diameter should first be determined. No fixed rule based on modern practice may be given for this ratio. In his "Modern Gas and Oil Engines," published some twenty years ago, Grover tried to show that on theoretical grounds, a ratio of 1½ should give the best results. An examination of the dimensions of a large number of the more modern stationary motors shows that this ratio is about the average, although ratios of from 1 to 2 may be found.

In small two-cycle motors, it is quite usual to find the bore and stroke the same, while in the very small power units the bore quite frequently exceeds the stroke.

It is also necessary to decide at what speed the engine is to be run. In stationary practice on small powers a piston speed of 600 feet per minute is quite common for horizontal engines, and of from 700 to 800 feet per minute for vertical engines. In extremely large power units a piston speed of 700 feet per minute for horizontal engines is quite common.

For small motors the speed is usually influenced by the purpose for which the engine is designed. For the average high speed marine engines for small boats and speed boats, a speed of 800 to 1,200 r. p. m. is quite usual, the latter speed for racers and the former for pleasure boats. For cruisers and work boats the speed range is between 400 and 600 r. p. m. although high speed motors have been very successful in cruisers in spite of a prejudice against their use for this purpose.

For automobile motors the usual speed is 1,800 r. p. m. and speeds of 3,000 to 4,000 are in use. Airplane motors are usually operated at about 1,200 and generally at speeds above 1,000 r. p. m.

The cylinder diameter and the speed of the engine having been determined, the balance of the dimensions of the engine are based almost entirely on these figures, although other factors enter more or less into the design of the several parts and will be discussed in their turn.

CHAPTER XX.

THE CYLINDER.

An average example of gas engine cylinder design is shown in Fig. 40. This particular type of cylinder construction is that employed on engines of 50 H. P. per cylinder and below for stationary service when the cylinder and cylinder head are cast separately. The reader will find, by referring to larger descriptive works on this subject, quite a wide variation in cylinder practice, both as to general form and proportions. Therefore the reader will not be able to take up any gas engine design he chances upon and find the dimensions exactly fitting the formulas here given. However, by observing the suggestions and following the equations in this chapter he will be able to secure a practical design which will be in good proportion.

In the proportions of the cylinder wall to the bore, designers differ widely. The reason for this difference is first that foundry practice varies, and the minimum cylinder wall that can be cast for a certain size cylinder depends somewhat on the skill of the foundryman. For this reason it will be found that writers on this subject will give the minimum thickness of the cylinder wall variously as ¼-inch to ⅝-inch. Now, as a matter of fact, foundry practice is by no means the limiting factor in this respect as in the casting itself the wall is much thicker than the finished cylinder. In small cylinders, where this factor is supposed to have most weight, modern tools will remove ¼-inch of metal with the same facility that they will ⅛-inch or less. As a matter of fact, in the modern aeroplane cylinder of as much as 4½ inches diameter, walls not over ⅛-inch thick are employed successfully. The limitations of the foundry do, however, influence the thickness of the water-jacket wall when it is cast as an integral part of the cylinder.

The basis of cylinder wall thickness is therefore strength, to which is added a certain amount to allow for *reboring without* reducing the factor of safety. In extremely large cylinders, wherein the resistance to pressure of the wall does not act as a simple shell, and the inner wall is under greater proportional strain than the outer, an additional allowance should be made.

The general formula for wall thickness is

$$t = \frac{PD}{2S}$$

Wherein t = thickness of wall in inches.
P = maximum pressure in pounds per sq. in.
D = cylinder bore in inches.
S = allowable safe stress.

For the average gas and gasoline engine the value of P is usually not over 320 lb. per sq. in. For larger engines using high compression this pressure may go as high as 450 lb. in rare instances. (See Chapter XVIII.) The value of S for cast iron is variously given by authorities from 1,500 to 4,000 lb. per sq. in.

An examination of quite a number of designs gives $t = .09\,D$ for average practice including an allowance for reboring.

The depth of the water-jacket, the radial measurement, may be found by the formula

$$j = .1\,D$$

Wherein j = the depth of the jacket.

For the average foundry the value of j should not be less than ½-inch, although the author has succeeded in securing good castings with j = ⅜-inch in special cases. Thinner water spaces than this give too large a percentage of bad castings to make them practical.

For the water jacket outer wall the practice of designers will be found to vary over a wide range. The author has used for engines of his own design a thickness one-half that of the cylinder wall or $.045\,D$. The limiting thickness for this wall is generally 3-16-inch, although by careful founding as thin a jacket wall as ⅛-inch may be employed in special cases.

The position of the counterbore should be such that with the piston at the inner dead center, the piston ring nearest the end of the piston should project into the counter-bore *from* one quarter to one third of the width of the *ring. It is* a mistake often made by designers to allow *merely the* end of the piston to overrun the cylinder bore.

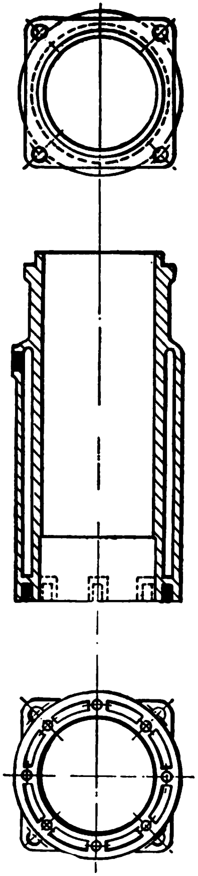

FIG. 40.

The object of the counter-bore is to avoid wearing a ridge at the end of the piston travel. This object is not fulfilled unless the ring overruns for the reason that on a properly designed piston that portion beyond the last ring should be smaller than the balance of the piston, and it is the ring that determines the end of the wear.

The length of the cylinder should be such that the piston overruns its open end. Except in special cases where the designer finds it advisable to place a ring at the open end, a trunk piston (open end type) may overrun the open end of the cylinder a distance equal to 20% of the length of the piston.

In order to determine the diameter of the cylinder head studs, the maximum pressure of the explosion, the number of studs and the safe stress of the material of which the studs are made must be known. The number of studs is seldom less than four. It is a good plan to limit the distance from center to center of the studs to six inches as a greater distance will allow the joint to spring open between them. Naturally, a very thick and stiff cylinder may make a greater distance than this permissible, but this distance has been found in practice to be a safe limit. An even number of studs is generally employed.

The following formula should be used for finding the diameter of the stud:

$$z^2 = \frac{D^2 P}{s\,n} \text{ or}$$

$$z = D\sqrt{\frac{P}{s\,n}}$$

Wherein D = the diameter of the cylinder in inches,
z = diameter of stud at root of thread,
s = safe stress in lb. per sq. in.,
P = maximum explosion pressure,
n = the number of studs.

The steel customarily employed for studs has a tensile strength of 60,000 lb. per sq. in. and with this steel a safe stress of 10,000 lb. per sq. in. may be employed. If there is any doubt about the value of P, it is generally safe to use 400 lb.

The flange by which the cylinder is attached to the frame of a horizontal engine or to the base or crank-case of a vertical engine is quite frequently made square when four studs or bolts are used and round when the number exceeds four. The sizes of the studs or bolts should be calculated from the same formula as for the cylinder head studs.

The thickness of the cylinder wall between the end of the water-jacket and the flange should be at least as great as water-jacketed portion of the cylinder. It is frequently made slightly larger in diameter. This may be of some advantage in horizontal engines to assist in carrying the weight of the overhung cylinder, but for vertical engines the author has found that a thickness of .09 D gives satisfactory results in practice. The thickness of the flange should be equal to the diameter of the bolt plus ⅛-inch.

On small engines of say 8-inch bore and below, the cylinder and the engine frame of horizontal engines are often cast in one piece. On very large bores and especially on double-acting engines the cylinder and the water-jacket are separate castings. The length of the water-jacket should be such that it will extend at least 10% of the stroke beyond the end of the piston when at the outer dead center, for four-cycle motors and 20% beyond for two-cycle motors. It is quite general practice in Europe to carry the water-jacket the full length of the cylinder, but good results are obtained with the shorter jacket. For extremely light designs such as on motors for racing cars, speed boats and aeroplanes, quite a number of engines have been built with water-jackets that do not extend to the full length of the piston travel. While fair results have been obtained with short jackets on four-cycle engines, the author believes that better results are assured by the longer jackets. Several engines of the two-cycle type have failed miserably because of water-jackets extending only to the top of the exhaust ports. The sizes of the intake and the outlet water pipes depend upon the method of cooling, whether by forced circulation, water directly from the mains, thermo-siphon systems, radiators, etc., as well as the number of cylinders. *For the average* engine, where the circulation is taken *care of by a pump* or the water is admitted directly from

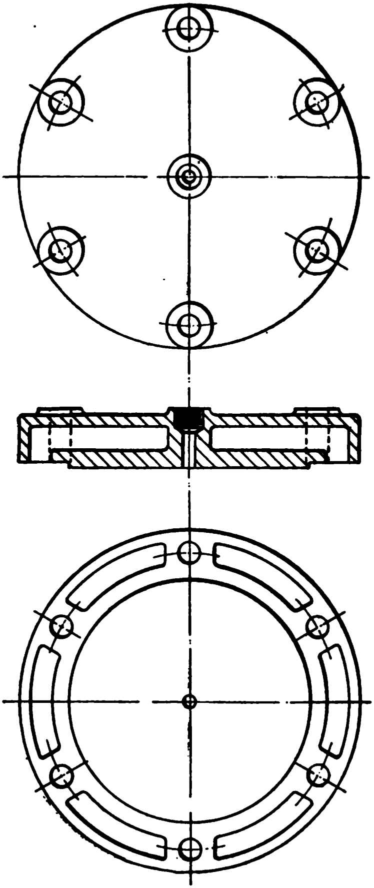

FIG. 41.

the water mains, the inlet and the outlet water pipes will be ample if .15 times the diameter of the cylinder. For thermo-siphon circulation in stationary engines using a large tank, and operating for the greater part of the time on partial load this size will be found satisfactory. But for automobiles using this system or for stationary work where the engine is heavily loaded, thermo-siphon circulation requires more capacity and the water pipes should be from 25 to 30 per cent. of the diameter of the cylinder. Or letting d represent the diameter of the water pipe, then

$d = .15\ D$ for forced circulation
$d = .25\ D$ to $.30\ D$ for thermo-siphon.

The cylinder head, when it contains no valve or other mechanism is made as shown in Fig. 41. The usual formulas for the strength of flat plates do not apply in this case for the reason that the head is strengthened by the jacket. There are always more or less bars across the jacket space similar to the central boss in the figure. Experience has shown that for small heads below 10 inches diameter, the inner wall of the head may safely be made the same thickness as the cylinder wall and for large heads or for heads of spherical contour 1.2 times the cylinder wall.

Fig. 42 shows a cylinder with head of partially spherical contour, with the head and the cylinder cast in one piece, in order that the reader may get a better idea of this style of construction. The proportions are shown as ratios to the cylinder diameter. Particular attention is called to the countersink in the center of the head to avoid pocketing the points of the spark plug when it is placed in the head. The two dead center positions of the top of the piston (the closed end) are shown in order to illustrate first the position of the end ring with relation to the counterbore, and second the position of the end of the piston at the outer center with relation to the end of the water-jacket.

The materials available for cylinders are unfortunately very few. Comparatively hard, close grained gray iron castings are most generally used. Semi-steel, gray iron with a small percentage of boiler punchings, is much used for automobile cylinders. Low carbon steel cylinders, bored from a solid hammered bar, are used to a con-

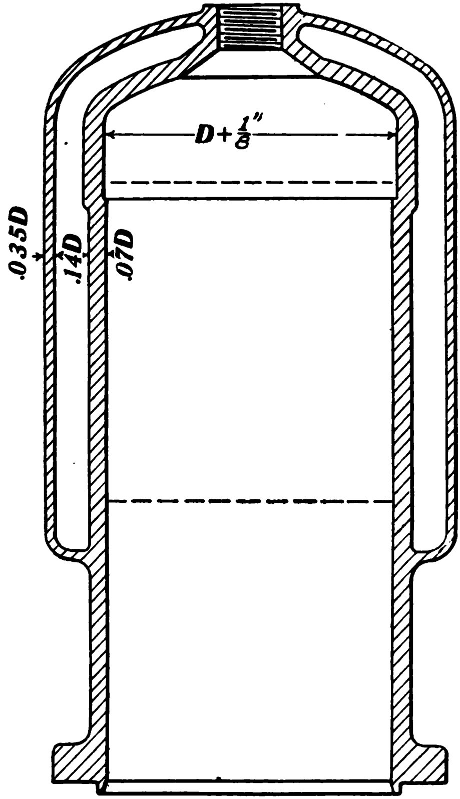

FIG. 42.

siderable extent for aeroplane motors in France. The author has used for cylinders of aeroplane and light marine engines an alloy of aluminum which has given excellent results in practice. Cast steel has been tried to some extent, but has been found to distort in use. Brass or more strictly speaking, *bronze* has been employed with indifferent results, and even wrought iron gas pipe has been used.

In the machining of gas engine cylinders, the greatest of pains should be taken to secure a straight and a perfectly true cylindrical hole. Aging of the castings after rough machining and before finishing is the best way to insure permanency of the work. Annealing after roughing answers this purpose in a way, but tests have shown that it reduces the tensile strength of the iron. Reaming, lapping and grinding are various methods employed for the final operation on the finished bore. Grinding is by far the most accurate, and gives much the best finish, but it is practical only on the smaller bores and is employed mostly for cylinders six inches or less in diameter. The counterbore of a small cylinder with integral heads may be cast to finished size, and the finish extend only to the edge of the counterbore as indicated in Fig. 42. Finishing the surface of the counterbore very smooth is supposed to avoid adherence of carbon and it is effective perhaps to a limited extent, but the author believes that with a carefully graphited core, a sufficiently smooth surface may be obtained.

The following points should be carefully looked after in the design and construction of cylinders. No pocket in the water spaces higher than the water outlet should be permitted. Such a space permits the accumulation of steam and interferes seriously with the water circulation. When designing cylinders to be cast "en bloc" (two or more cylinders in one casting) do not join the cylinder walls of two adjacent cylinders so that no water can pass between them. The result of integral walls is distortion of the bore and consequent leakage past the piston after the engine has become warm. Be sure to have all parts of the combustion space, no matter how small, water-jacketed. This applies especially to the valves and the valve boxes. An unevenly cooled cylinder will distort and *cause trouble. Both* the exhaust and the *inlet* valves should *be jacketed. Avoid* heavy blocks of metal where you can. *Where they cannot* be avoided be sure to call the foundryman's attention to them. Always test the water-jacket of a cylinder with at least 80 lb. water pressure both before machining and after rough machining and before finishing. A porous cylinder, if leaking only slightly, may be saved by rusting with a strong brine or a solution of sal ammoniac in water. Allow the solution to stand in the jackets from twelve to twenty-four hours.

CHAPTER XXI.

PISTONS, CONNECTING RODS AND CRANK-SHAFTS.

With few exceptions, gas engines of small powers are fitted with trunk pistons. The principal reason for this was the difficulty experienced by the earlier gas engine builders in satisfactorily packing piston rods of gas engines. Therefore the single-acting engine was the rule. Within the past decade, as gas engines of large powers came more and more into general use, the designers were compelled to make them double-acting in order to keep the over-all dimensions within reasonable limits. Careful experimenting with various forms of metallic packing and the adoption of water-cooled pistons and piston rods has overcome all difficulties with double-acting gas engines, and today they are quite common.

The general proportions of a trunk piston for a single-acting engine are shown in Fig. 43. The proportions are based on the cylinder diameter as a unit. All the necessary equations are shown on the drawing. For pistons smaller than 8 inches diameter the ribs *R* may be omitted, the inside of the head being made straight across, or crowned making it thinner at the center. The number of piston rings varies from three in motors of small bores, say five inches and under, to four and five in medium size bores and even as many as seven rings are used in some of the large single-acting engines of English make. A piston ring is sometimes placed near the open end of the piston, but in recent designs of the four-cycle type, this ring is now very seldom seen. It is, however, quite frequently used in two-cycle motors employing crank-case compression. The value of this ring in the piston of a two-cycle engine is very doubtful. The author has designed approximately one hundred engines of this type, has never used the ring near the open end of the piston, and has never missed it. It has been stated in defense of the end ring that after the piston wears it will not hold compression in the base. Careful observation of

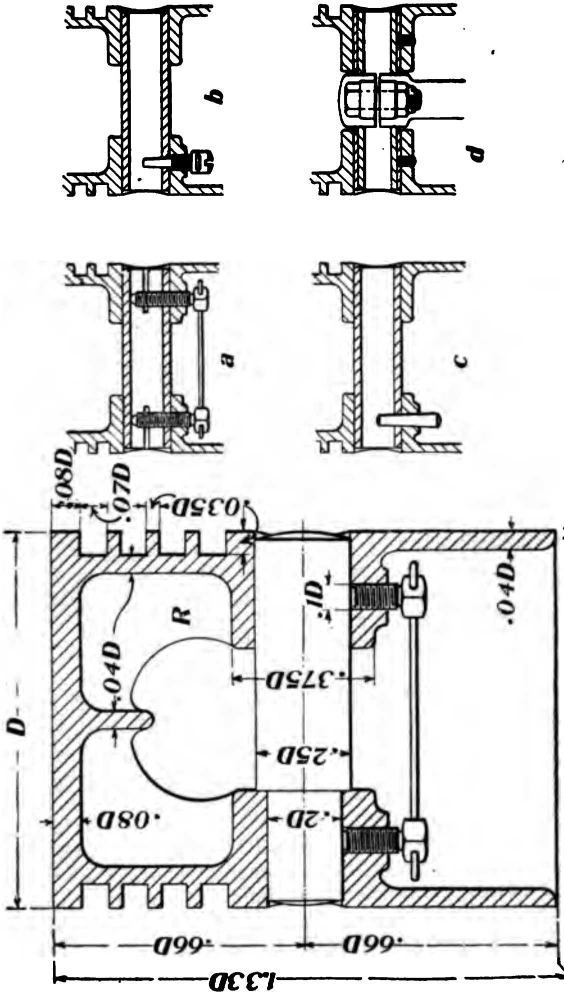

FIG. 43.

engines, after they have run for several years without this ring, shows no appreciable loss of power, and proves conclusively that a two-cycle will give good service with all the rings at the closed end of the piston.

In using the proportions given in Fig. 43, the designer should be careful to observe the practical limits and not make parts of the piston for small bores too light for practical service. For example, the wall of the open end should not be less than 3-16-inch, especially where the piston is not bored. For pistons of airplane engines or other engines of very light construction, the wall may be ⅛-inch, but in that case the inside of the piston should be machined, and a short distance back from the open end the wall should be 3-16-inch or the wall will be cracked in handling.

The piston pin may be hollow and made of steel tubing. The size of the hole may be two-thirds the outer diameter of the pin. The method of fastening the pin in place with set screws, requires some device to prevent the screws from turning and coming out. In any engine with a closed base, if a set-screw comes out it is quite sure to be caught between the rod and the wall of the base, and knock a hole through it. Therefore the method of locking the screw should be a certain one. It is very important that the piston pin be held not only from turning, but from moving endwise as well. Should the pin project from the side of the piston it will make a groove in the wall of the cylinder and ruin the cylinder.

Several methods of locking the pin are shown in the small drawings. At *a* is shown a very good method that may be used in connection with a hollow pin. The pin is split for a short distance from the end, and tapped for the set screw through one side only. The set screw is then screwed against the wall of the pin opposite the tapped hole. This spreads the end of the pin and tightens it in the piston so there is no danger of the pin knocking. The hole in the piston through which the screw is inserted prevents the pin from moving endwise. A wire passed through holes in the head of each screw and bent over, will effectually lock the set-screws and prevent them from turning. Do not set these screws too tight or they will distort the piston.

The piston pin in drawing *b* is held in place by means of a set-screw with a long taper point, and in one end only of the pin. The friction of the long taper will generally prevent the screw from unscrewing in use, but the design at *a* is the more certain in its security.

The simple taper pin shown in drawing *c* in one end of the piston pin has the advantage of security and ease of removal, but unless the piston pin itself be tapered the fit is not easily made a snug one.

In drawing *d* is shown a piston pin held fast in the end of the rod and allowed to move in the piston as a bearing. This form of construction, while used to quite an extent on small engines, is not generally considered good practice by gas engine designers. The objection to it is that when the hole in the piston wears oval it is more difficult to repair satisfactorily than a worn bushing in the end of a connecting rod. The claim of its advocates that it is more easily lubricated than when the pin turns in the rod, is not borne out in practice.

The piston ring is one of the most important parts of a gas engine, and one of the most frequently neglected, especially by the inexperienced constructor. Piston ring iron should be dense and free from imperfections of any kind, and as hard as can be turned in the lathe. One of the best tests of a piston ring is to break it and note the fracture. If a very light gray, nearly white, and of very fine grain, the metal is generally good. Poor ring material is nearly black in the fracture, and the grain has the appearance of coke.

To test the spring of piston rings, pull the ends apart about a half inch for a four inch ring, and if the ends come back together again without taking a permanent set, or do not take a set when the ends are sprung apart sidewise, the ring is generally a good one.

For high grade automobile and marine engines the practice of cutting piston rings from a sleeve or "pot" is no longer in favor. The latest practice is the use of a separate casting for each ring with very little finish. The rapid cooling of this small casting gives it the fine grain so much desired in a piston ring, and good spring.

A piston ring should be finished with the sides parallel and the periphery of the ring square with the sides. The *small* rings for automobile and other small engines are *usually* ground on the sides on a magnetic chuck. It is the practice of quite a number of factories to grind the periphery of the rings as well. But no matter how carefully the ring is ground, it is sure to be distorted more or less in springing the ring over the end of the piston to place it in the groove. The ring will therefore vary slightly from a true circle when in place. This variation is generally very slight and a few rings will not show any. What little there is can soon be worn out of a turned ring after a few hours' run. The ground ring is glazed in grinding and does not wear so rapidly as a turned ring and it takes some time to seat perfectly against the wall of the cylinder. For this reason, at least one prominent specialist in piston rings for automobile engines has abandoned grinding the periphery and finishes it with a very fine cut.

The general practice in ring design calls for an eccentric ring, thinner radially at the cut than at the opposite side. This design is for the purpose of making the ring press evenly against the wall of the piston at all points. A ring to be *theoretically* perfect in this regard would taper to nothing at the cut. As this is not practical, the proportions used are those shown in Fig. 44.

The character of the slot in the ring has been the basis of much argument among designers. The lapped joint, shown at *b* has quite a few adherents. It is a good joint, not difficult to make, but is weak at *k* and especially in two-cycle engines where the ring has to pass across the ports is apt to let the ring catch into the port if broken.

The diagonal slot shown at *c* appears to give good results in practice, and if the steady pin of the ring is made as shown it has all the advantages of the lap joint.

The use of a pin to keep the ring from turning around the piston is imperative in two-cycle motors where the rings pass across the ports. Should the opening in the ring come opposite the port, it will allow the end of the ring to enter the port and damage both the cylinder and the piston.

Pistons for engines of the double-acting type are much like those of a steam engine for small four-cycle motors, but for large engines of either type they are hollow and water-cooled. In double-acting two-cycle engines *the length of the piston* is the length of the stroke minus *the width of the* exhaust port. Fig. 45 gives the general

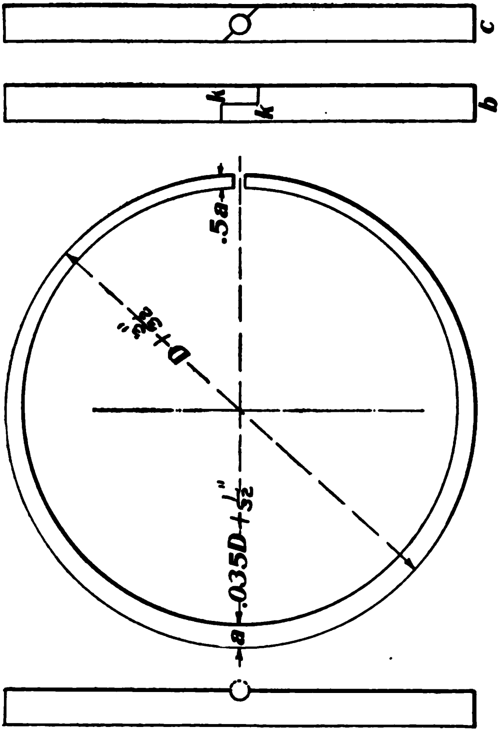

FIG. 44.

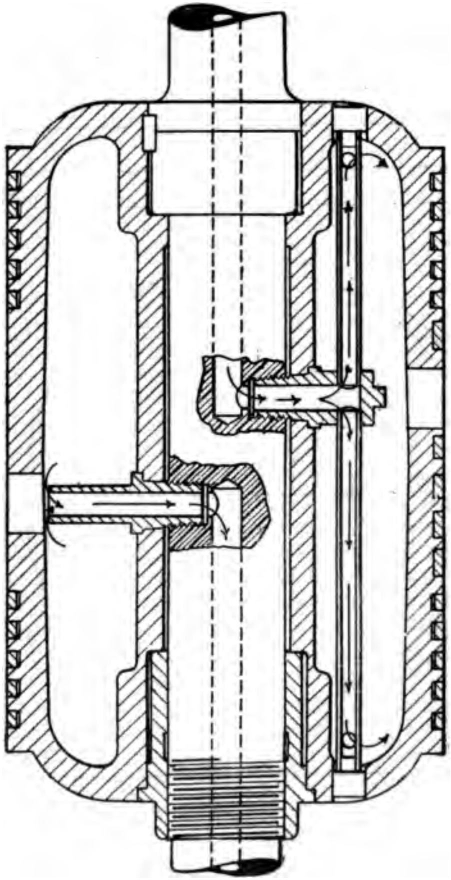

FIG. 45.

proportions of a large piston for a double-acting two-cycle engine and the arrows show the path of the cooling water.

The connecting rod of the gas engine is customarily made in one of two general forms, both of which are illustrated in Fig. 46. The rod *I* is that usually found in small engines such as the automobile type, and the rod *II* shows the marine type quite generally used in stationary engines, although the strap and wedge form is found occasionally. The cross-section of the rod may be circular, slab (rectangular) or I-beam in form. It is made slightly tapering, the larger end being that nearest the crank-pin.

The taper of the average connecting rod is about ⅛-inch per foot of length between centers, although no fixed rule is followed. The mean diameter of a rod of circular cross-section may be found by the following formula:

$$r = .035\sqrt{Dl\sqrt{m}}$$

Wherein r = the mean diameter of the rod;
D = the diameter of the cylinder;
l = the length of the rod between centers in inches;
m = the maximum or explosion pressure.

In the case of rod with a rectangular cross section, it is necessary to first determine the ratio of the height h with relation to the width b, the height being the measurement in a direction at a right angle to the crank-shaft. Assuming the ratio of h to b as 1.85, a good average value, the formula becomes

$$h = .0385\sqrt{Dl\sqrt{m}}.$$

If the rod is of I-beam section, as shown at *III*, Fig. 46, the proportions may be as follows for good effect:

$b = .5h$, $b' = .15h$, $h' = .6h$. With these proportions,
$h = .0371\sqrt{Dl\sqrt{m}}$.

Should the cross-section of the rod be a hollow cylinder, *i. e.*, tubular, we may assume that the average case would have the diameter of the hole in the rod .9 the outside diameter and the value of d based on this assumption would be

$d = .0457\sqrt{Dl\sqrt{m}}$.

The designer should not forget that resistance to

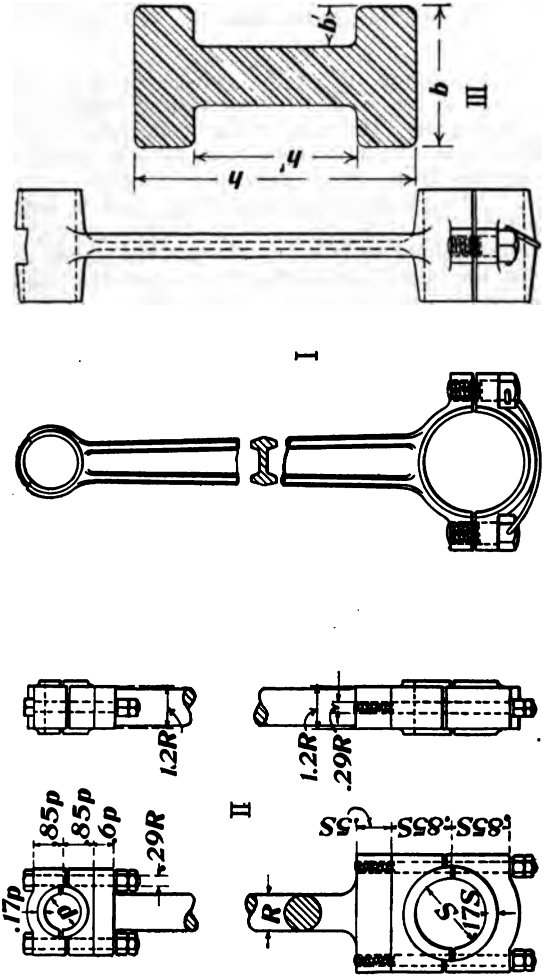

FIG. 46.

bending, which is the only factor considered in the above formulas, is not the only thing to look out for. He should also take into consideration the resistance of the rod to compression. Quite frequently resistance to compression outweighs resistance to bending. It should be remembered that the resistance to bending depends upon the modulus of elasticity, which is very nearly the same for all steels. The resistance to compression, on the other hand, varies with the material.

In figuring the size of a rod to resist compression it is usually safe to assume that the resistance of the material to compression is about two-thirds the resistance to tension. Therefore if the tensile strength of the material is 60,000 lb. per sq. in., the compressive strength may be taken as 40,000 lb. When the resistance to compression is being computed, the designer should consider the smallest section of the rod and not the mean area. Furthermore, the area of the rod at its smallest section should be carefully and not roughly calculated, especially when the rod is small.

The formula for resistance to compression is

$$a = \frac{.7854D^2m}{f}$$

Wherein a = area smallest section of the rod,
D = diameter of cylinder,
m = maximum pressure,
f = safe load for the material in the rod.

Taking a factor of safety of 6, the safe load for the various materials usually employed for connecting rods will be about as follows:

MATERIAL	$f=$
Phosphor Bronze	5,000
Manganese Bronze	6,600
Cast Steel	6,200
Low Carbon Steel	6,600
High Carbon Steel	8,800
Crucible Vanadium Steel (Heat Treated)	13,500

By low carbon steel is meant steels with about 20 point carbon and by high carbon steel is meant those of about 40 point carbon.

The general proportions of the average connecting *rod are shown* in Fig. 46. In a rod for a double-acting engine it may be necessary to increase the diameter of the bolts or studs to hold the caps for the bearings. In this type of engine each bolt should be computed to withstand two-thirds the tensile strain upon the rod.

The crank-shaft of a gas engine has to withstand, for the same average power, a greater initial strain than the shaft of a steam engine. The maximum pressure is usually four times the average or mean effective pressure. The crank-shaft of a gas engine must therefore be much larger than that for a steam engine of the same power.

To find the diameter of the shaft, the diameter of the cylinder and the maximum pressure is taken as the basis of computation. The following formula gives the diameter of the shaft:

$$s = k \sqrt[3]{mLD^2}.$$

Wherein

s = diameter of the shaft in inches,

k = a constant depending on the material,

L = length of stroke in inches,

m and D as for the rod.

The value of k is .056 for wrought iron, .052 for mild steel and .047 for 40 pt. carbon steel which is that usually specified for small high speed motors, except where special alloy steels are employed. These special steels for crank-shafts will be discussed fully in the chapter on aeroplane motor design.

This formula may give larger shafts than are found in some engines, but it gives safe values. The diameter of the crankpin varies from the diameter of the shaft to 1¼ times the diameter of the shaft. It should not be less, for average engines, than .4D.

The general proportions of a crankshaft that follows conservative practice are shown in Fig. 47. The proportions are based on the diameter of the shaft as a unit. The length of the crankpin should not be less than the diameter, and it should be so designed that the pressure per square inch of projected area should not exceed 400 lbs. per square inch. From this rule the following formula for the pin is derived:

$$f = \frac{AP}{400q}$$

Wherein

q = *the diameter* of the pin,

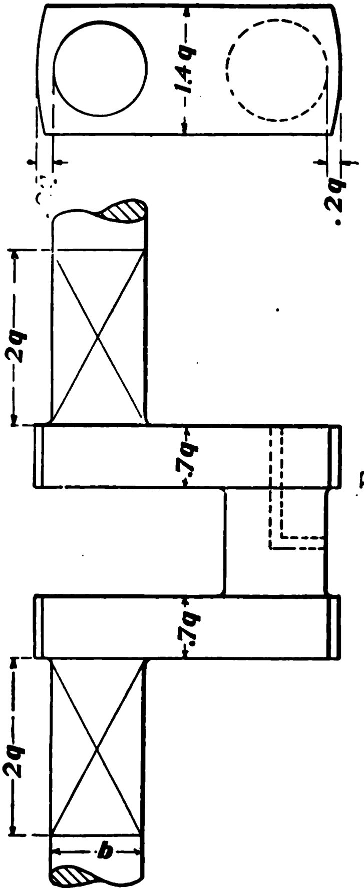

FIG. 47.

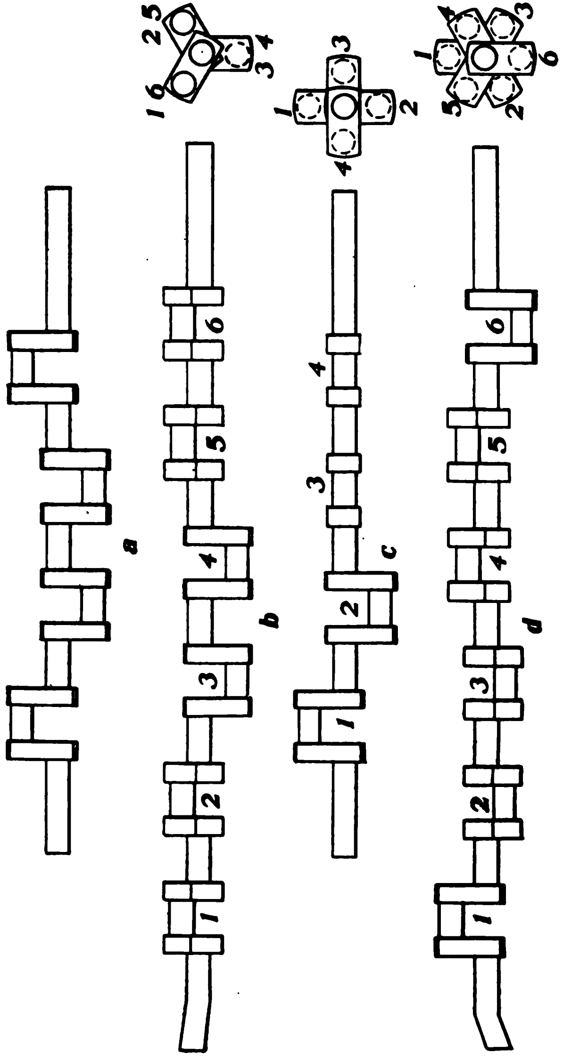

FIG. 47II.

f = the length of the pin,
A = area of the piston,
P = mean effective pressure.

The length of the pin should be further determined by the following formula:

$$f = \frac{APR}{400{,}000}$$

R = r. p. m.

Both methods of calculating the length of the pin should be employed and the longer of the two chosen. This is particularly true of high-speed motors in which the latter formula will usually be found to give the larger value.

For multiple cylinder engines the diameter of the shaft nearest the end where the power is taken off should be designed to withstand the maximum load given by all the cylinders, calculating from a carefully laid out diagram showing the sums of the pressures at various points in the crank circle.

The arrangement of the crankpins on a four cylinder four-cycle motor should be as shown in Fig. 47IIa, for a six, as in Fig. 47IIb, and for a four cylinder two-cycle as in Fig. 47IIc, while for a six-cylinder two-cycle the arrangement should be as in Fig. 47IId.

The formula for the diameter of a hollow shaft is as follows:

$$d^3 = \frac{d_1^4 - d_2^4}{d_1}$$

Wherein

d = the diameter of the shaft found by the formula,
d_1 = outside diameter of hollow shaft,
d_2 = inside diameter of hollow shaft.

For example, the inside diameter of a shaft 2½ inches outside diameter which is equivalent to a solid shaft 1¾ inches diameter is very close to 2¼ inches. In the example given the hollow shaft weighs about 39% of the weight of the solid shaft.

CHAPTER XXII.

VALVES.

As the general principles of design, involved in the mushroom type, apply to valves of other forms, the following discussion will be confined to this form. The proportions of the passages both for the admission of the charge and the emission of the exhaust should be founded upon a speed at which the gases will flow through the passages with a very low friction. For this reason the areas of these passages depend both on the area of the cylinder and the speed of the piston. Quite frequently these areas are made smaller than they should be, with consequent wire-drawing and loss of power.

The basis of the design of the inlet passages is to so design them that the speed of the gases at any point shall not exceed 100 ft. per second. For the exhaust passages, the limit of speed is taken at 85 ft. per second. The allowance for the exhaust is an arbitrary one and is, for convenience, based on the assumption that the gases in the cylinder at the time of opening the valve are at atmospheric pressure. As a matter of fact the pressure of the gases at the moment of release ranges between 30 and 40 lbs. per sq. in. and the lower limit of speed for the exhaust gases is chosen for this reason.

The designer should remember that the piston speed of gas engines varies considerably, even for engines of the same power when made by different manufacturers, and therefore it should always be taken into consideration when proportioning the passages. Do not overlook the fact that to design the inlet and the exhaust pipes according to the above basis and then to choke the column of gas by a small valve opening or a constricted inlet port, is not very good practice. It may seem strange to the experienced designer to call attention to this point, but the author has found more engines at fault in this one particular than in any other way. In fact, I have found *valves of ample* area with inlet and exhaust pipes one-fourth the area of the valves. These mistakes were not always made by inexperienced men, but by designers who should have known better.

Based upon the above assumptions of limiting speeds, the following formulas will give the designer short methods of determining the sizes of the passages:

Let S = the speed of the piston in feet per minute.
A = the area of the cylinder;
a = the area of the inlet passages;
a^1 = the area of the exhaust passages;
D = the diameter of the cylinder;
d = the diameter of the inlet passage;
d^1 = the diameter of the exhaust passage;
L = the length of the stroke in inches;
R. = the r. p. m. of the crank-shaft. Then

$$a = \frac{AS}{6{,}000} \text{ or } a = \frac{ARL}{36{,}000} \text{ for the inlet;}$$

$$a^1 = \frac{AS}{5{,}100} \text{ or } a^1 = \frac{ARL}{30{,}600} \text{ for the exhaust;}$$

$$d = .00527D\sqrt{RL} \text{ for the inlet;}$$

$$d^1 = .00572D\sqrt{RL} \text{ for the exhaust.}$$

When the piston speed of the engine is that frequently employed for stationary engines (600 ft. p. m.) these formulas may be further simplified to

a = .1 A,
a^1 = .118A.

and for passages of circular cross-section at the same piston speed we have

d = .316 D,
d^1 = .343 D.

When the piston speed is above 600 ft. per min., the values will be found from the following table, and for speeds between, the next highest speed may be taken without serious error.

Piston Speed, Feet per min.	a =	a^1 =	d =	d^1 =
700	.117A	.138A	.342D	.371D
800	.133A	.157A	.365D	.396D
900	.150A	.177A	.387D	.421D
1,000	.167A	.197A	.409D	.444D
1,100	.183A	.216A	.428D	.465D
1,200	.200A	.236A	.447D	.486D

The designer should use the formula best suited to his requirements, and he will find it of assistance to check his work by means of another formula.

Valves for gas and gasoline engines are usually made of steel for the inlet valve, quite frequently of drop forgings. Ordinary carbon steel will not stand the heat of the exhaust and soon becomes pitted if used for this purpose. Cast iron has been found the most suitable material for the exhaust valve as it will neither pit or distort. Exhaust valves are usually made of cast iron heads attached to a steel stem. Nickel steel has been found to stand up well for exhaust valves and 3½% nickel steel is frequently employed for this purpose made with the head of nickel steel and the stem of carbon steel welded to it. While it is not necessary to use this material for the intake valves, for the convenience of having all alike and interchangeable, they are usually made the same size and of the combined steel for small motors.

On large motors, nothing seems to quite take the place of the cast iron head. Even in special cases for small motors the cast iron valve will stand up where the nickel steel valve falls down. A good example of this is in the Wright aeroplane motor, on which the head of the cylinder is not water-cooled. For this purpose, there was finally developed a valve with a cast-iron head cast on a steel stem.

The objection to the cast-iron head is the difficulty of attaching it permanently to the steel stem. In addition to casting the head around the end of the stem, several other methods have proven satisfactory. One plan is to thread the end of the stem to be inserted in the head, and tap the hole in the head smaller than the thread on the stem. The cast-iron blank is then heated and the stem screwed in quickly by means of a crank handle or by holding the stem in the chuck of a lathe and starting up the lathe the moment that the stem is started into the head. The end of the stem protruding through the head is then riveted over without destroying the center in the stem and the whole valve finished afterward.

The usual proportions of a gas engine valve are shown in Fig. 48. Valve I is an inlet valve, which may be made much lighter at the rim than the exhaust valve. Valve E, shows an example of an exhaust valve. The projection T,

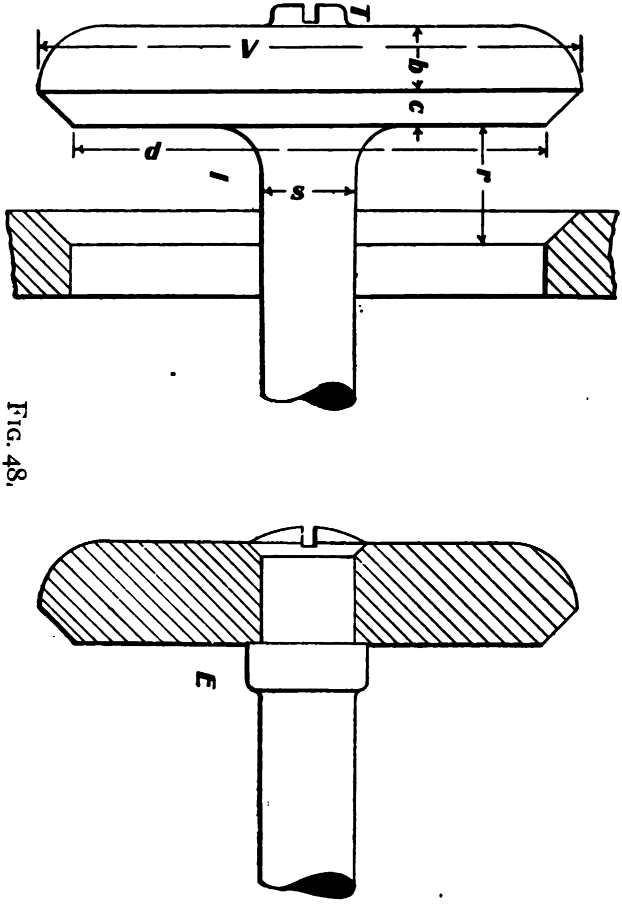

FIG. 48.

should be provided with a screw-driver slot for convenience in grinding the valve.

The following proportions will give good satisfaction in practice, provided good materials are used:

$V = 1.14\ d$
$b = .14\ d$
$c = .07\ d$
$s = .2\ d$ for short stems,
$s = .23\ d$ for long stems,
$r =$ the lift of the valve, not less than $.25\ d$.

For high speed engines where the lift given by this formula is for any reason objectionable, the area of the opening under the valve is retained by increasing the diameter to correspond to the reduction in the lift. The area of the opening under the rim of the valve is $3.1416\ rd$. In this formula r represents the lift and not the radius.

In Figs. 49, 50 and 51 are shown various arrangements of the inlet and the exhaust valve, and the proportions of the various parts of the valve housing in terms of the cylinder diameter and parts of the valve. Fig. 49 illustrates an arrangement with both valves in the cylinder head Fig. 50 shows them side by side, either as they would appear in a separate valve chest or when in the side of an L-shaped cylinder. Fig. 51 shows the arrangement of the valves on opposite sides of the cylinder, when the latter is made in what is called the T-head form.

One of the most important features of valve design after determining the proper size to use, is the timing of their opening and closing. In the earlier designs, and especially in stationary engine practice, it was formerly the custom to open the inlet valve at the begining of the suction stroke and to close it at the end of the suction stroke. The exhaust valve was opened about ten per cent of the piston travel from the end of the expansion stroke and closed just at the end of the exhaust stroke.

If the inlet valve is closed at the end of the suction stroke, the cylinder cannot become completely filled with the fresh charge. Unless the inlet valve be exceptionally large and the movement of the piston very slow, there is a considerable vacuum in the cylinder at end of the stroke Furthermore, the column of gas entering the cylinder has a certain amount of momentum which tends to keep it rushing through the inlet passages even after the suction

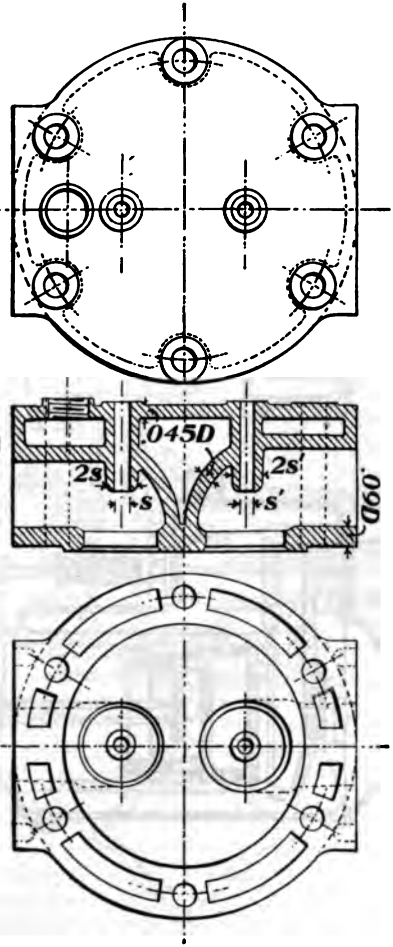

FIG. 49.

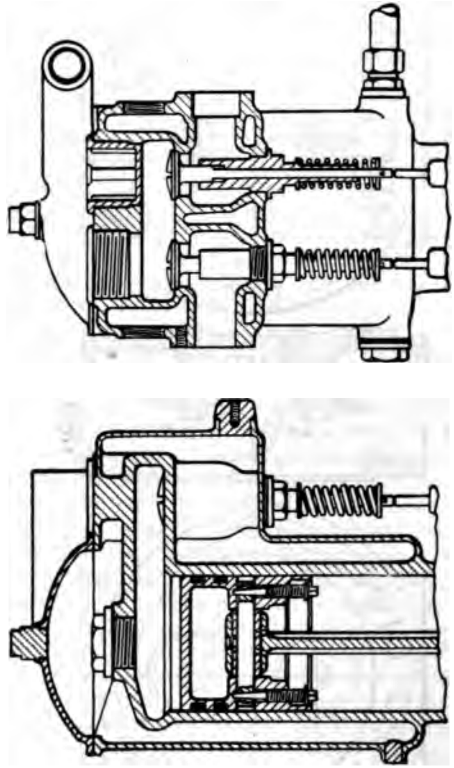

FIG. 50.

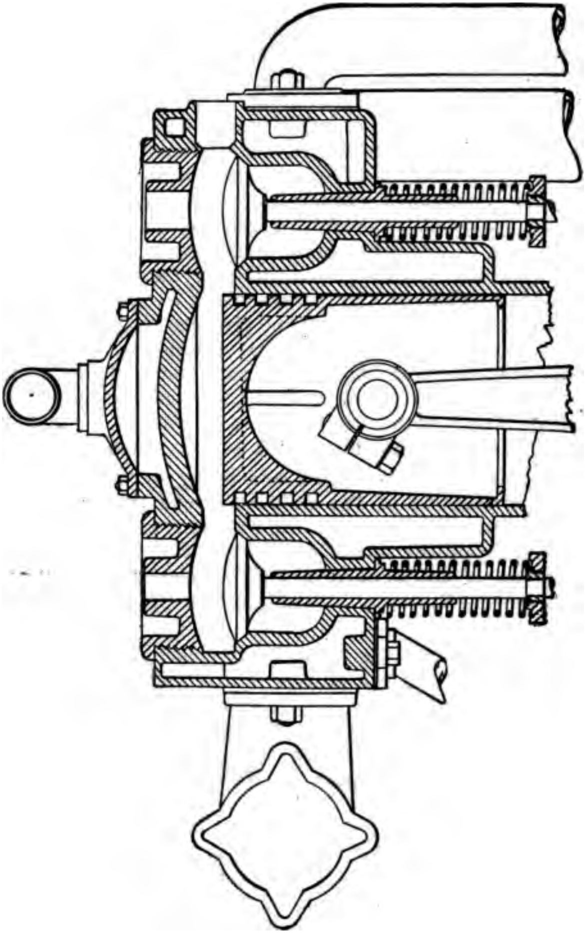

FIG. 51.

has ceased. For this reason the cylinder will be more completely filled if the inlet valve be allowed to remain open for a short time after the end of the suction stroke and during a portion of the compression stroke. It should be remembered that during the first part of the compression stroke the piston is moving with comparative slowness and therefore the crankpin can move through a considerable angle for a small movement of the piston. There is a point in the compression stroke beyond which the charge would be driven back through the valve. This point is reached earlier in the stroke at low speed than at high speed. Therefore a good method to determine this point is, to find what point is the latest that the valve may be closed without getting a blow back or reversal of the air column at the lowest speed at which it is desired to run the engine. It is obvious that for racing boats and automobiles, this point should be selected at the normal speed of the motor, as a back draft at low speed is of small moment and as the engine is invariably throttled at low speed in this class of power, a valve set for high speed will not blow back at low speed.

When the exhaust gases are being driven from the cylinder at the end of the exhaust stroke they have considerable momentum and, if the exhaust valve is allowed to remain open for small portion of the suction stroke, the opening of the inlet valve being delayed, the cylinder will be more effectually cleared of the products of combustion.

In determining the time in the cycle most suitable for the opening and the closing of the valves, the modern designer measures in degrees of crank-pin travel and not in fractions of piston travel. The reason for this is that the speed of the crank-pin is uniform while that of the piston varies in every part of the stroke. The time of opening and closing of the valves on various types of four-cycle engines is given in the following diagrams.

In Diagram *A*, Fig. 52, is shown the average setting of the valves of four-cycle automobile engines as given by Messrs. Kessler and Lewis in *The American Machinist* of Oct. 13, 1910. As in steam engine practice, the opening or closing of a valve before the piston reaches the dead *center is known as lead* and after the dead center as *lag*. *During the first* or suction stroke, the inlet has a lag

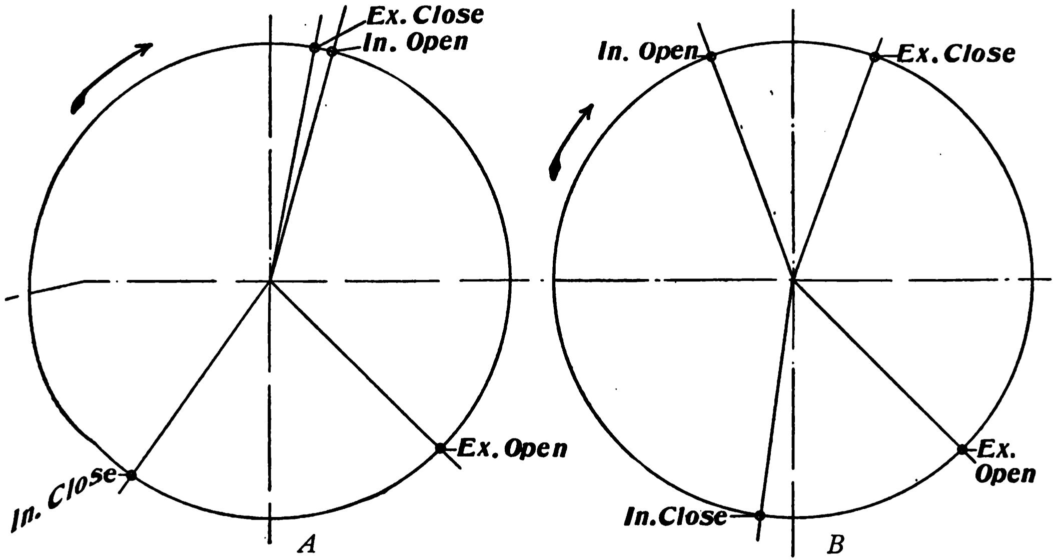
Ex. Close
In. Open
Ex. Open
In. Close
A
In. Open
Ex. Close
Ex.
Open
In.Close
B

FIG. 52.

in opening of 14.7° and a lag in closing of 35.4°. The opening of the exhaust valve has a lead of 45.3° and it closes with a lag of 10.3°. The closing of the exhaust valve leads the opening of the inlet valve by 4.4°. The author has seen as much as 51° lead for the opening of the exhaust and as much as 45° lag in the closing of the inlet valve. In T-head motors (Fig. 51) the inlet valve may be opened before the exhaust valve closes, the angle between the opening of the inlet and the closing of the exhaust being from 2° to 4° according to the size and speed of the motor. The larger the motor and hence the greater the distance across the cylinder from the inlet to the exhaust valve and the higher the speed, the greater may be the angle.

Diagram *B* Fig. 52 shows the valve setting of a large stationary gas engine designed for scavenging, *i. e.*, clearing the cylinder of all the burned gas. At the end of the exhaust stroke there usually remains a small portion of the products of combustion that the piston will not drive out. By careful design of the exhaust passages, so they are free and have considerable length before they reach the atmosphere, the momentum of the column of gas will tend to carry with it the contents of the compression space. For this purpose, the inlet opening is given a lead of 20° and the exhaust a lag of an equal amount so that the two valves are both open during 40° of crankpin travel. The gas valve is opened just before the exhaust valve closes or at the time of closing the exhaust. The length of the exhaust pipe must be from 40 to 65 feet and all sharp bends avoided. The arrangement of the exhaust and inlet ports as well as the shape of the compression space must be such that the entering air will have a free sweep across the end of the cylinder.

The final test of the proper setting of the valves on any engine is by means of the indicator diagram. It is best for the designer to thus carefully test the valve setting on any new design of engine before making it final. The main object to be kept in mind is the securing of the highest volumetric efficiency. This efficiency is the ratio of the actual volume of the charge taken into the cylinder to the piston displacement.

Let $V.E.$ = volumetric efficiency,
v = volume of new charge, in cu. in.,
D = diameter of cylinder, inches,
L = stroke in inches.

$$\text{Then } V.E. = \frac{v}{.7854\,D^2L}$$

This should not be confused with the apparent volumetric efficiency. The difference between the real and the apparent value is due to the expansion of the fresh charge when entering the cylinder because the interior of the cylinder is hotter than the surrounding air. The measurement of the real volumetric efficiency is obtained by measuring the charge as it enters the cylinder. The apparent value is that which would be obtained from the indicator diagram, by measuring the distance from the end of the stroke, that the compression curve crosses the atmospheric line and comparing this with the length of the stroke. Suppose for example, that the compression curve crossed the atmospheric line at .7 of the piston travel from the end of the stroke, then the apparent volumetric efficiency would be 70%.

The reader will see, from the definition that it is quite possible to secure over 100% volumetric efficiency, by filling the entire cylinder, the piston displacement and the clearance, with fresh mixture.

In the design of valve springs, there are two factors to be considered. The springs must have sufficient tension to bring the valve to its seat with sufficient speed to make the cam roller follow the cam at all times during its opening and with mechanical valves the spring must be sufficient to hold the valve to its seat during such periods of the cycle that there is a vacuum in the cylinder due to throttling or the operation of the governor. The time of the return of the valve to its seat must be determined from the cam form.

Let W = the weight of the valve,
r = the lift, in inches,
t = the time of closure in seconds.
f = the pressure in pounds (spring tension).

$$\text{Then } f = .00518\,\frac{Wr}{t^2}$$

To f should be added the weight of the valve, when *the stem of* the valve is vertical and closes upward.

or the weight of the valve should be subtracted when it closes downward. To the pressure f should be added sufficient to overcome friction. Gueldner recommends that f be from .7 to 1 lb. per square inch of valve for automatic valves.

For mechanical valves the same authority advises $f=kd^2$.

Wherein d = the diameter of the valve in inches,
k = 4.35 to 7.07.

To resist the vacuum within the cylinder the value of k in the above formula should be .7854 v, where v is the vacuum in pounds per square inch.

The selection of the proper spring requires some judgment and observation of springs on existing engines. The scale of a spring is the pressure in pounds required to deflect the spring one inch.

Let D = the diameter of the coil at the center of the spring wire.

w = the diameter of the wire,
n = the number of coils.
S = the scale of the spring.

$$\text{Then } S = 1{,}440{,}000\frac{w^4}{nD^3}$$

For automatic valves n should be from 5 to 8, and for mechanical valves from 6 to 15, choosing the larger number when the circumstances will permit.

The reason for choosing the greater number of coils is that the longer the spring the less will be the change in the tension for the same movement. The diameter D of the coil is usually about .75 d (where d = the diameter of the valve) on small and moderate sized engines and about .5d on very large engines. The diameter w of the wire should be chosen from the safe load on the spring. The safe load F should be obtained by means of the following formula:

$$F = K\frac{w^3}{D} \text{ or } w = \sqrt[3]{\frac{FD}{K}}$$

The value of K varies with the size of the wire as *follows:*

w	K
Less than ¼″	400,000
¼″	275,000
⅜″	235,000
½″	196,000

To use the formula, it is necessary to select approximately the size of the wire, and then test the selection by means of the formula. For wire between the figures given, use the next smaller value of K. Placing F equal to f, the necessary pressure required on the valve, is the first step in the use of the formula.

Having chosen the size of the spring, it must be designed of such a length that when in place on the engine with the valve on its seat it requires a pressure equal to f to bring it within the space designed for it. Suppose that the pressure required is 20 lb. and that the scale of the spring is 10 lb. The spring must be compressed 2″ to secure the required pressure when it is in place. Therefore if the space designed for the spring is 5″ long, the original length of the spring should be 7″. The space between the coils when the spring is compressed should be not less than one-half the diameter of the wire.

The best way to designate the size of a spring in the drawing is to give the inside diameter, the size of the wire, the number of turns per inch and either the total number of turns in the spring or the length over all. Valve springs are usually designed with flat ends, and this also should be specified in the drawing for the information of the spring maker. Always let the drawing show the *free* length and not the length of the spring after compression.

It will usually be found that for the exhaust valve, heavier springs are employed on stationary engines than on the same engine for the inlet valve. The reason is that no allowance is made for suction opening the inlet valve. This is necessarily the case with an automatic inlet valve. On this type of valve, it is necessary to supply a catch to hold the valve to its seat when the governor acts on the exhaust valve to hold it open during the idle stroke.

The mechanism for opening the valves takes various forms according to the ideas of the designers and the particular result desired. These devices are so numerous that it is impracticable to treat them all here and the principal forms are described in Chapter V. The levers of the valve operating mechanism are similar in principles of design to levers on other machines. The cam, however, is a part of the mechanism which is frequently puzzling to the designer. Numerous attempts to design a cam that will give a certain movement, such as uniform acceleration, have generally resulted in a noisy mechanism. The reason is that the spring is not able to make the valve, and the mechanism between it and the cam follow the surface.

Practical experience has shown that with a roller on the cam the most satisfactory form is the flat sided cam shown in Fig. 53. The cam is made up of the circular portion *C* concentric to the cam shaft and the projection or eccentric portion *P*. When there is no separate spring to hold the push rod or other mechanism so that the roller bears against the cam, the part *C* is turned a little smaller than the circle shown by the dotted line which is the circle which would be described by the roller at its periphery when the valve is down on its seat and there is no lost motion between the valve stem and the device carrying the roller. In laying out the concentric portion it should be remembered that the valve does not begin to open until the point *p*, where the line meets the dotted circle, has come in contact with the roller. That portion of the revolution of the cam during which the valve is to remain open should, therefore be laid off on the dotted circle, and not, as the author has seen it done, on the concentric outline of the cam. The angle *P* is laid out according to the angle during which the valve is to remain open, this angle being half the crank angle for the same operation. The radius of the arc *n k* is that of the dotted circle plus the lift *r*. After laying off the angle *P*, draw the lines *jk* and *mn* passing through the points *q* and *p* and tangent to the circular outline of the cam. Slightly round the corners at *n* and *k*, and the cam outline is complete.

Recent practice calls for independent springs to return the push rod or other cam-operating mechanism. In *that case the roller* is allowed to run on the concentric *portion of the* cam and the necessary clearance or lost *motion is between* the push rod and the end of the valve

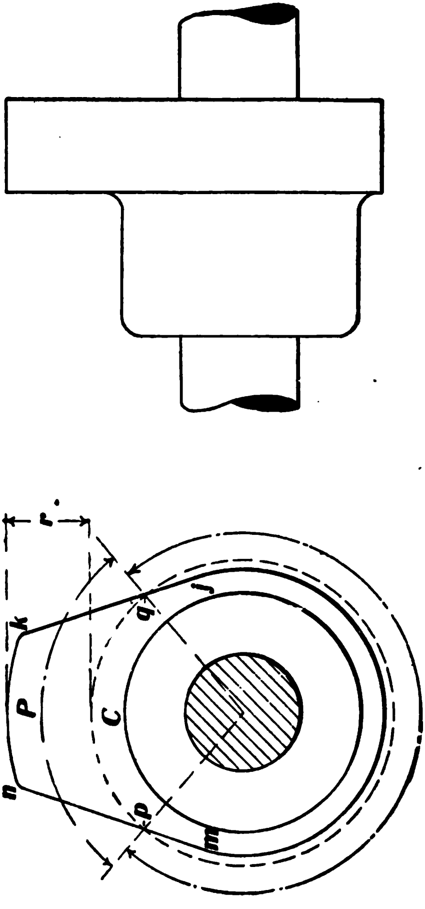

FIG. 53.

stem. This plan is followed when the movement of the cam roller is transmitted to the valve stem by means of levers. The return of the levers is accomplished by means of a spring separate from that of the valve stem. This spring relieves the valve spring of the extra labor of handling the operating medium and eliminates the blow of the cam on the roller. The shock of the light blow of the end of the push rod or tappet rod on the end of the valve stem may be cushioned by means of a small piece of hard fiber. (See Fig. 10).

The designer should keep in mind the weak points of the roller push rod when designing this type of mechanism. These parts are the roller shaft and the guide pin to keep the push rod from turning. Quite frequently, the arrangement for adjusting the clearance between the end of the push rod and the end of the valve stem is flimsily constructed. Two forms of push rod are shown in Figs. 10 and 11.

The troublesome features of the roller type of push rod are eliminated by use of the push rod with a mushroom head as illustrated in Fig. 11. This form of push rod is made in one piece of tool steel carefully hardened and, if both the cams and the heads of the push rods are properly tempered, it will give fully as lasting service as the roller type. It will readily be seen that it has none of the weaknesses of that type. The author's experience with this device extends over a period of several years of hard service on tractor motors.

In Fig. 54 is shown the method of laying out the cam for this type of push rod. The easiest way is to lay out the path of a point on a roller type using the plan described in Fig. 53, and taking up each successive step in the movement of the push rod, draw the cam outline tangent to the head of the rod, as illustrated at *a*, *b* and *c*. It will be found that in most instances the inlet cam will be brought very nearly to a point and the exhaust cam will have a short portion concentric to the low circle of the cam.

It should be noted that this form of push rod requires no guide as it can be left free to turn. It usually does *turn slightly at* each revolution of the cam, thoroughly distributing the wear. This mechanism should be so enclosed that it will be subject to flooded lubrication.

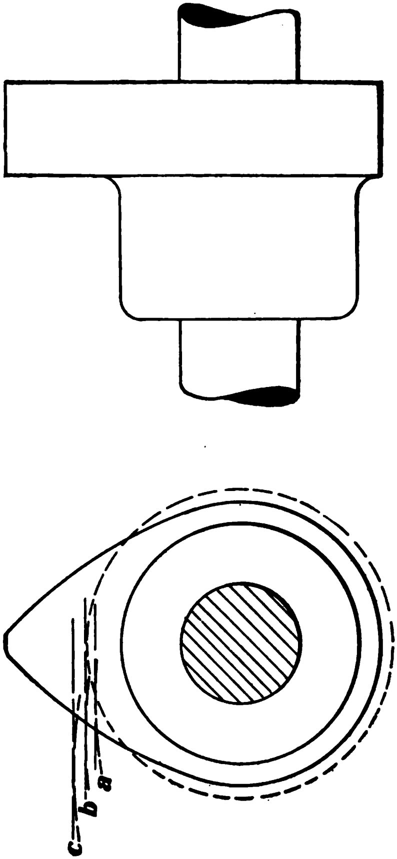

FIG. 54.

CHAPTER XXIII.

IGNITERS.

The general subject of ignition devices has been fully discussed in Chapters IX and X. Their design requires the following of a few general principles, and the avoiding of mistakes quite frequently made by the novice. To understand thoroughly the underlying principles of the make-and-break system the reader should study thoroughly the discussion in Chapter IX.

The design of a low-tension ignition system involves principally the securing of good electrical contact between the electrodes and their quick separation at the moment when the spark is required. There should also be a sufficient time of contact to secure the required saturation of the core of the spark coil. The latter, however, depends more on adjustment than design.

Referring first to Fig. 22, Chapter IX, the movable electrode *p* is drawn away from the stationary electrode *f* by the snapping of the spring *s*. While this gives a comparatively rapid separation of the points of contact, if the points could be knocked apart by hitting *p* with a hammer, the separation would be much more rapid, with a corresponding increase in the intensity of the spark.

An igniter in which the principle of the hammer blow is employed is illustrated in connection with the snap magneto system in Fig. 23. The principle involved is easier to understand from Fig. 55. In this figure the spring *s*, used to bring the contacts *c* and *c* together, is much weaker than the hammer spring *S*. Just before the time of ignition, the cam *C* pushes the collar *K* out of contact with the arm *a* on the outer end of movable electrode *m* and the spring *s* brings the contacts together. At the proper time for the spark, the cam releases the rod carrying the collar *K* and it strikes a hard blow on the end of the arm *a*, sharply separating the contacts. While the mechanism shown in the figure has been in successful use, quite a number of variations of the device, all involving the same principles, are in use.

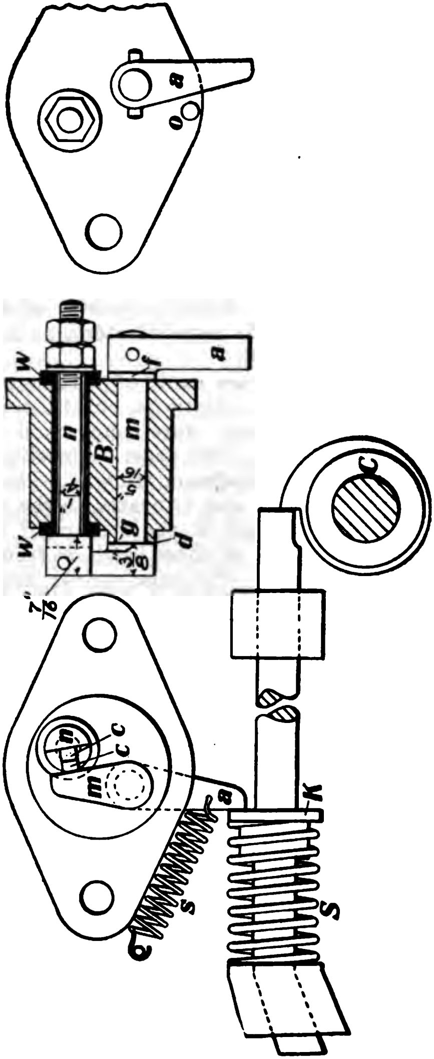

FIG. 55.

As the same size igniter will answer for very nearly any size of engine, the dimensions of the parts are given in inches and not in formulas. Brass is the favorite material for the body *B* of the igniter, while a good grade of machinery steel is satisfactory for the electrodes *m* and *n*. The stationary electrode *n* should be insulated from the body with mica washers at *w w*, and a mica tube should surround the stem. This tube need not fill the entire space between the washers as, if it is a little short, it will effectually keep the stem clear of the body. There should be two nuts on the outer end of the stem, one to hold the electrode in place and the other to clamp the wire. Some designers use a screw in the end of the stem to hold the wire in a hole drilled crosswise of the stem. This form of binding post is unsatisfactory, as it cuts the wire and does not hold well.

The contact points *c c* are made of various materials. Platinum, platinum-iridium, platinum substitutes such as "meteor wire" and "cassaloy," pure copper and even tool steel have been employed. The most expensive and the best wearing material is platinum iridium. This is supplied in the form of beads, and is brazed to the electrode, the latter usually being sent to the makers of the alloy for this purpose. The size of these beads is from ⅛-inch to 3-16-inch in diameter. Platinum points are usually made from ⅛-inch wire, and either brazed or riveted in. Meteor wire and cassaloy should be 3-16-inch wire and turned to ⅛-inch making a stem as shown in the figure, and the end of the stem is then riveted in place. Copper points should be ¼-inch diameter driven into a taper reamed hole and riveted over.

The movable electrode *m* should be ground to a gas tight bearing at *d* and a light spring or spring washer used at *f* to keep the joint to its seat. The stem of the electrode may be relieved in the center allowing it to bear at the ends. It should always move freely. Do not fail to round the corners at *g* into a small fillet, counter-sinking the seat to clear the fillet. Neglect of this precaution is apt to result in the head breaking off the stem.

The time of contact between the points *c c* is obtained by the adjustment of the collar *K* and the stop pin *o*. The time of contact depends upon the length of the coil. For the five-inch coil usually employed upon a high speed engine a contact of one two-hundredth of a second is sufficient.

This would mean that the points should be in contact for 18° of the crank circle when the speed of the engine is 600 r. p. m. For the 10-inch coil used on slow speed engines, double the time of contact should be employed or 18° of crank movement, at 300 r. p. m. It is best to use one or two degrees more to be on the safe side.

In caring for a make-and-break igniter, be sure to keep the parts free and clean the igniter occasionally. The dirtier the fuel, the oftener should the igniter be cleaned. See that the contact points meet squarely, so they have a broad surface of contact. If they are corroded or fouled or meet with a small surface of contact, the current will not pass freely and the ignition will be unsatisfactory.

So locate the igniter that the gases have a free flow across it when entering the cylinder. This will help to keep the points cool, clean and in the body of good mixture. The points should project into the combustion space and not into a pocket. Do not carry this principle too far and have them project far into the center of the compression space as this will permit them to heat easily.

The design of jump-spark apparatus, so far as it concerns the engine man is chiefly one of location of the spark plug and the construction of the timing device. The design of spark coils and spark plugs is so specialized that it is merely a question of choice.

The location of the spark plug in a four-cycle motor is generally in the cap over the inlet valve or in the center of the cylinder head. Extra spark plugs are generally placed in the cap of the exhaust valve. They should always be so located that the gas has a free sweep across them and not in a depression or pocket. If the points project into the compression space ⅛-inch, it is usually sufficient. If the wall is thick use a long plug or countersink the space around the plug.

The location of the spark plug in a two-cycle motor should be in the center of the cylinder head or on the side nearest the deflecting plate. Never place the spark plug in a two-cycle on the same side of the cylinder as the exhaust port.

In large engines, the use of two or more spark plugs at different points in the compression space has shown good results, as they reduce the period of inflammation with a consequent increase in the mean effective pressure. In small motors for automobiles with the valves located on opposite sides of the cylinder (T-head type) a certain magneto company has shown an increase of as high as 20% in the delivered power of the motor by placing plugs over both valves. These plugs are supplied with current by a specially wound high-tension magneto so they both fire at the same instant, the gaps being in series. While no such result is claimed for this system with the L-head, or in a two-cycle motor, the author has obtained an increase of over 5% in the speed of a two-cycle motor with two plugs, when driving a screw propeller.

In the test referred to, the motor was fitted with two independent systems of ignition, one a magneto and the other a coil and distributor with one plug in the center of the cylinder head and the other in the top of the head at the edge of the bore, on the same side as the by-pass. Either system would operate the motor at 900 r. p. m., while with both systems in operation the speed increased to 950 r. p. m. The same results were obtained with a high-tension distributor system on one set of plugs in each case, while the other set on one motor was served with a high-tension magneto, and on the other motor with a low-tension magneto and vibrator coils. As the power required to drive a propeller increases as the cube of the speed, the increase in power was approximately 17%. The motor in each case was a three-port four-cylinder 4½-inch diameter and 5-inch stroke.

The design of a timer for a jump spark coil requires a moving contact in metallic connection with the engine (grounded) and suitable stationary contacts, one for each cylinder. The time during which electrical contact for each coil should endure is about the same as for a five-inch coil on a make-and-break igniter, or one two-hundredth second. This duration of contact varies with the spark coil, but for motors running at a speed of 1,000 or 1,200 r. p. m. the duration of the contact should be 34° of crank movement. This means 34° on the timer of a two-cycle motor and 17° on the timer of a four-cycle motor. More than this gives waste of current and less, unsatisfactory ignition at high speeds.

Several forms of timing and high-tension distributing systems are illustrated and described in Chapter X. A

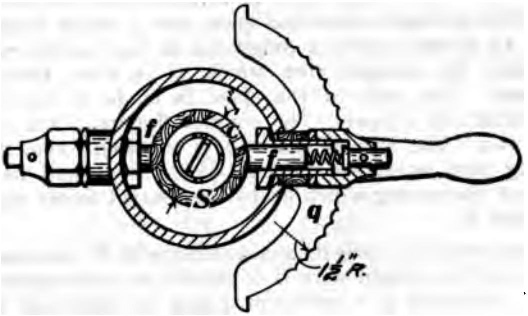

FIG. 56.

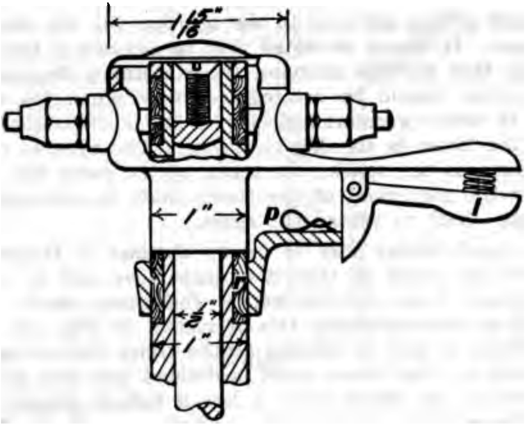

FIG. 57.

simple form of jump-spark timer for vibrator coils is shown in Figs. 56, 57 with the principal dimensions given. The insulation of the spool *S* which carries the moving contact *c*, is hard vulcanized fiber, and is made from tubing. As there is quite a difference in the quality of this material, the purpose for which it is used should be specified. The body of the spool is made of brass and the contact of copper. The contact fingers *f* are yellow brass rod. The wire is inserted in the small hole in the binding post as shown by the dotted lines and held in place by the spring which keeps the contact finger against the spool *S*.

For small two-cycle motors which are to be reversed on the spark, the ground wire is connected to a spring binding post *p*, attached to a quadrant *q* and the quadrant is insulated from the engine by means of the fiber collar *r*. Pressing the latch *l*, breaks the circuit, and allows the motor to slow down. By keeping the timer advanced and releasing the latch at the right moment ignition takes place too early for the momentum of the flywheel to carry the piston over the center and the force of the explosion drives the motor backward, starting it in a reverse direction. Then the lever is swung to the other side of the quadrant giving advance to the ignition for the changed direction. It should be noted that to advance a timer by turning that portion carrying the stationary fingers, the timer lever should be rotated clockwise when the timer shaft is rotating counter-clockwise and counter-clockwise when the timer is turning clockwise. The reverse movement retards the spark. In other words, move the lever against the movement of the timer shaft to advance and with the shaft to retard the spark.

A much better way to fix up a timer is fasten the body of the timer so that it cannot move and to rotate the contact point in relation to the timer shaft. One method of accomplishing this is shown in Fig. 58. The timer body is held in the top of the timer column by the set screw *s*. The timer shaft is divided and that portion attached to the timer spool *S* has a helical groove *g* in which travels the pin *p*, fast in a collar *c*. In the lower end of *c* is a slot *k* which slides on the pin *x* in the upper end of the timer shaft. When the collar c is raised by means of the fingers *f*, the pin p turns the spool shaft.

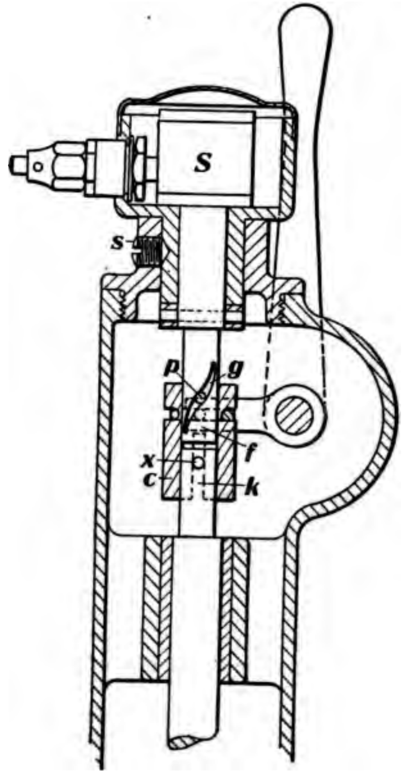

FIG. 58.

In a system of this type, the spool is rotated in the same direction as the timer shaft revolves to advance the spark and in the opposite direction to retard the spark.

To advance the spark on a magneto the system described in Chapter X in connection with Fig. 27, shows a lever to turn the circuit breaker for the low tension winding. This system has the disadvantage, that the farther away from the point of maximum voltage the circuit is broken, the weaker will be the spark at the plug. Therefore to secure the best volume of spark at all positions of spark advance, it is best to rotate the armature of the magneto with relation to the position of the driving gears, by some system of the type illustrated in Fig. 58. This same advance may be accomplished by driving the magneto by means of a pair of helical gears and sliding one gear lengthwise.

The advance of the spark for best effect will be found to vary with the fuel employed, the mixture, the compression and the speed of the engine. The range of the advance for high speed engines such as are employed for automobiles, boats and aeroplanes should be from 30° to 45° of the crank movement. The better the quality of the spark, the less the range that is required.

The choice of a spark plug is very often one of the most puzzling points to the user of the jump spark. Quite a number of so-called trouble-proof plugs are quite the opposite of what is claimed for them. In order to show the variations in spark plug design, four leading examples are shown in Fig. 59. Plug *A* has a porcelain insulator through the center of which passes a nickel wire *c*. On the upper end of the porcelain is spun the brass cap *k* which is very close to the upper end of *c*. The porcelain is held in a steel body by the nut *n*, bearing against a shoulder on the porcelain and making a gas tight joint by means of the gasket *g*. At the lower edge of the body is the nickel wire hook *h*, which is bent toward the wire *c* forming the gap at *x* across which the spark is formed. In this plug the spark jumps a short gap between the upper end of *c* and the cap *k* as well as at *x*. The reason for not fastening *c* to *k* is to allow for the difference in the expansion of *c* and the porcelain. If some provision of this sort is not made, the porcelain is liable to crack.

At *B* is shown another form of plug in which steatite

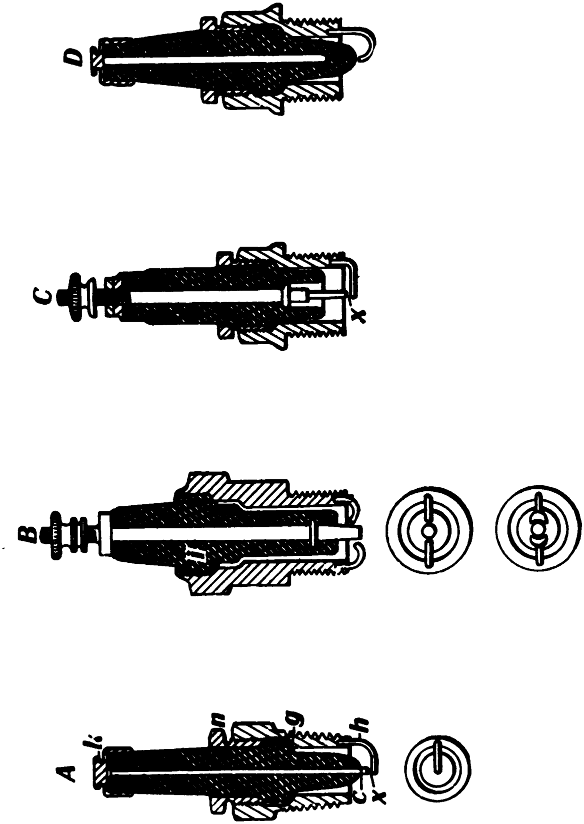

Fig. 59.

(soap-stone) is employed for the insulator *i*. Two hooks are shown on this plug and sometimes three are provided. A later form has the ends of the hooks flattened as shown in the lower end view. The steatite insulator in this form is spun into place and is not removable. It does not appear to collect soot as does porcelain.

At *C* is shown a plug with a petticoat type of porcelain. The idea of this form is that the distance on the surface of the porcelain between the body and the center wire is too great for a coating of soot to short circuit. In other words, the resistance of the soot is greater over this space than the resistance of the spark gap at *x*.

The plug shown at *D* has the lower end of the center wire made from an exceedingly small platinum wire. The spark to this wire is so small that its intensity will overcome any fouling.

The plug *A* is a very good form that will meet widely varying conditions. If well made, it gives good satisfaction, but poorly made plugs of this type foul easily, especially when running light and using an excess of oil. To cheapen the plug the makers often use moulded porcelains, which are frequently poorly glazed. The cap *k* is sometimes omitted and a nut is threaded on the outer end of the wire *c*, with no provision made for expansion. Cheap porcelains crack and flake. If not properly glazed the spark will pass through the porcelain instead of across the gap at *x*. The best porcelain is made from imported clay, hand turned, carefully baked and heavily glazed.

Plug *B* gives excellent service. The hooks shown on the upper view are excellent for heavy loads and in hot places where the gas from the intake does not pass over them as in the center of the cylinder head of a two-cycle engine. They are not so good for slow speed as the plugs with crescent points. The latter, however, should be placed in the path of the fresh gases, for if allowed to heat they incline to bridge and will sometimes burn off at the points of the crescent.

Plug *C* has the same objection as the crescent points of plug *B* in that it will not give satisfaction in hot places. The cause of the trouble is different, however. The long exposed end of the center wire heats up and causes the charge to fire prematurely. It does not work well in some two-cycle engines for this reason.

Plug *D* gives good results if the inner end of the porcelain is ground off occasionally. On a fairly heavy current the end of the fine wire appears to burn off gradually and to increase the gap at *x*.

Two styles of thread for attaching the plug in place are in use. One is the half-inch pipe thread and the other the A. L. A. M. pattern with ⅞-inch diameter and 18 threads per inch. On the latter type the plug has a shoulder and is made gas-tight by means of a gasket. The ½-inch pattern is shown on plug *A* and the A. L. A. M. on the other plugs.

In Europe a pattern similar to the A. L. A. M. is employed with a metric thread.

Of the two styles the shoulder plug is much preferable to the taper type thread. On the latter there seems to be no standard, and plugs from different makers cannot be depended upon to be the same size on the thread. In fact plugs from the same lot may vary as much as two threads in the distance they will screw into the cylinder. The shoulder plug always enters the cylinder the same distance, makes a neater appearance than the taper plug for this reason and the same position of the points inside the cylinder gives more even timing of the ignition. The effect of the position of the spark in the cylinder is that the ignition is slower with a spark in a recess.

CHAPTER XXIV.

FRAMES AND BASES.

The frame or base of a gas engine is that part which carries the crank-shaft bearings and which supports the engine upon its foundation, either directly or through the medium of a sub-base. Besides acting as a support, the frame takes the entire thrust of the explosion between the bearings and the cylinder. In engines of the vertical type, such as those employed for automobiles and boats, as well as for certain forms of horizontal engines in which the crank and the connecting rod are entirely enclosed, this part of the engine is usually known as the *base* and in some instances (particularly in two-cycle practice) as the *crank-case*. In stationary horizontal engines the term *frame* is more generally employed.

In the case of the enclosed type of base, wherein the metal is symmetrically disposed about the shaft bearing, the strain is one of simple tension. But in the usual design as found in the frames of the horizontal type of stationary engine there is, in addition to the tensile, a bending strain of considerable moment, usually fully as great as the tensile strain and frequently from three to four times the tensile. In some designs there are shearing strains of more or less importance. All of these must be added to the tensile strain when determining the proper proportions of the frame section to withstand the load. In addition to the bending and shearing strains, there are certain designs in which there is a strain tending to spread apart the arms supporting the bearings, strains on the bearing caps, and a strain tending to break out that portion of the frame beyond the bearing cap.

It is in the frame design and in the computation of its strength that we find the inexperienced designer the most neglectful, and inclined to work by "Rule of Thumb." In fact, the earlier designers of steam engines (See Thurston, "A Manual of the Steam Engine," Vol. II, page 122) rarely made computations in the design of a frame.

Works on steam engine design and even many works on gas engine design are singularly lacking in this respect.

In the smaller engines, such as those employed for automobiles and boats, the exigencies of the foundry require castings of a thickness that usually gives a wall amply large for strength. As the sizes of these engines increase, the importance of computation increases and it is well to compute the strength of a proposed section in any case. For small iron bases, it is customary to make the wall not less than ¼-inch thick, although in special cases the author has succeeded in securing satisfactory castings with 3-16-inch wall and even as thin as 5-32-inch. In aluminum, the customary wall is ¼-inch, but as low as ⅛-inch has been employed with satisfactory results.

It is in the design of frames for large stationary engines of the horizontal type that the computations of the various strains assume the greatest importance, and as these designs are an excellent guide to almost all general forms, they will be treated in detail.

The majority of gas engines are built of the center-crank type, with a bearing for the shaft on both sides of the crank-pin, the side bearing crank with but a single crank arm (the Corliss type) being used but rarely. It has been tried in several instances by European designers only to be finally abandoned as unfit for gas engines. In spite of this fact, several prominent American builders of high-powered gas engines are successfully employing the side crank.

Several examples of engine frame design are given in Figs. 60, 61 and 62, showing current practice, and to illustrate the principles involved in computing the strains in the various parts.

One of the most important features of frame design is the location of the shaft bearing with relation to the neutral axis of the frame section. The neutral axis is located at the center of gravity of the cross-section of the frame taken on the line *A-B*, Fig. 60-I. The greater the distance *a* between the axis of the cylinder and the neutral axis of the frame, the greater will be the bending strain. The total load of the explosion upon the piston multiplied by the distance *a*, is the bending moment and *a* *is the moment* arm. The stress caused on the outer fiber *of the frame* by this bending moment must be added to the

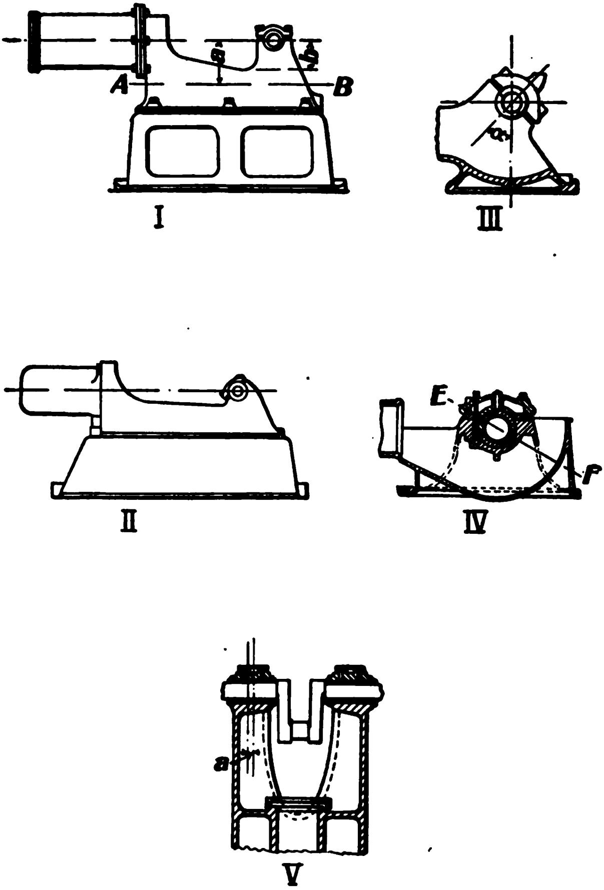

FIG. 60.

tensile stress, in order to determine the *total load* or stress on the metal of the frame.

In Fig. 60 I, which is taken from one of the earlier designs, there is a bending moment *a* times the load on the piston to be considered as affecting the projecting part of the casting carrying the bearing. In this part of the casting there is a bending stress only and not a tensile stress. As it is customary to consider one side of the frame at a time in computing the strains, the load carried by one side will be one-half the total.

In Fig. 60 II, is shown a frame with a bearing having the parting at an angle slanting downward toward the cylinder. In this form it is necessary to have the line of pressure above the neutral axis, giving a considerable bending moment, on account of the length of the moment arm.

In Fig. 60 III, the parting of the bearing is given a downward slope away from the cylinder. This design will permit the lessening of the moment arm, as the top edge of the frame can be brought closer to the center of pressure, thus raising the neutral axis. This form of design brings into play a considerable strain on the cap and bolts of the bearing, which must be reckoned with and the caps and bolts designed accordingly. The strain on the cap is $mA\sin a$; where m is the maximum pressure of the explosion in lb. per sq. in., A is the area of the piston and a as shown in the figure.

Fig. 60 IV, shows a form of bearing quite widely used in modern horizontal engines. This style of bearing permits the top edge of the frame to be carried above the line of pressure and the neutral axis of the frame section to be brought very closely to the line of pressure if not coincident with it, thus reducing the bending moment to a minimum. The line *E F* on this bearing shows a possible line of fracture in any bearing of this type, and the strength of this part should be checked.

Fig. 60 V, shows another form of frame in an engine having the shaft above the piston in which the pressure has a tendency to spread the arms, bend them outward. In this style of frame the moment arm is that shown at *a*. There is a similar force to be reckoned with in the design of frames for the Corliss type of engine frame.

There is one point conclusively shown by the various

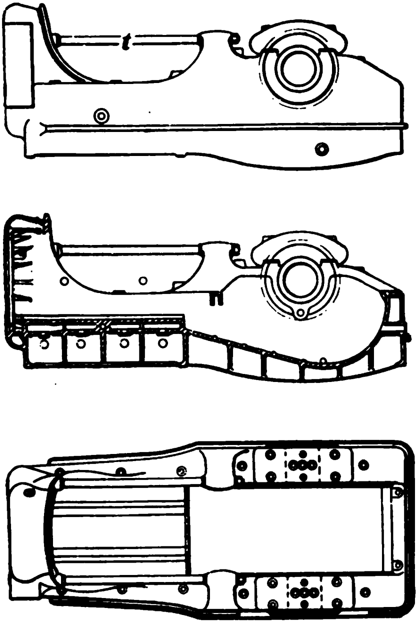

Fig. 61.

designs in Fig. 60 that will be more apparent as the computation of the bending moment is more fully considered. That is the importance of reducing the moment arm *a* to the smallest possible length in order to avoid an excessive amount of metal in the frame. As the size of the engine increases, this point impresses itself upon the designer, and it will be found that the reduction of *a* by as much as ⅛-inch has a marked effect upon the maximum fiber stress.

In the case of the vertical type of engine with an enclosed base, the neutral axis of the frame is in a plane passing through the center of the cylinder and the moment arm *a*, Fig. 60, is zero. In some of the later designs of large stationary engines with frames having the neutral axis below the center of pressure, a tie rod is used to balance the bending moment as shown at *t*, Fig. 61, from Gueldner's "Internal Combustion Engines." This frame is that of a 1,000 H. P. double-acting Nurnberg engine. The strength of the tie rod can be made such that it will take all or part of the bending strain. In the illustration the rods are designed to withstand the entire thrust of the explosion.

Fig. 62 shows the proportions of an average design of frame as employed in modern stationary engines of large size. The proportions are based on the diameter of the cylinder as a unit, and will be found to give good results in practice. For large castings in gray iron Gueldner recommends that the total strain in pounds per square inch should not exceed 2,800. By following the proportions of the drawing and retaining the distance *b* of Fig. 60 within .2*D* this strain will not be exceeded, when the maximum pressure does not exceed 350 lb. per sq. in. By the examination of designs of well-known makers it will be found that the width of the base or lower part of the frame varies considerably. Usually the average will be found 3*D* and it sometimes goes above that figure. The height of the center of the shaft above the base is usually very close to *D*. From 28″ to 30″ is a convenient height above the floor for the operator, and on small engines a sub-base is employed.

In order to illustrate the manner of computing the strain there is shown in Fig. 63 a frame for an engine with a cylinder diameter of 20″ with the proportions based

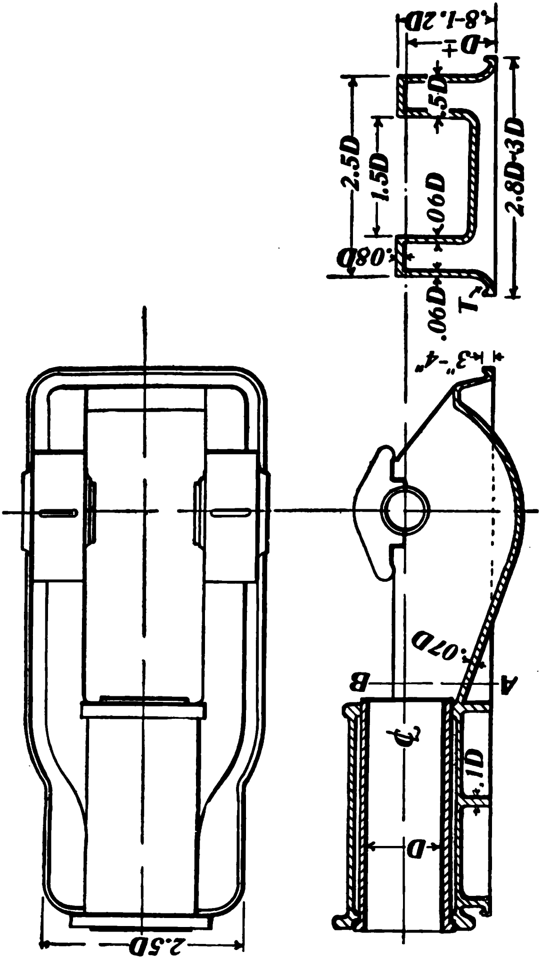

FIG. 62.

on the formulas of Fig. 62. The height of the top of the frame is equal to the cylinder diameter and the center line of the cylinder is taken also equal to *D*. The width at the bottom is equal 3*D*. As the frame is symmetrical about the vertical plane passing through the center of the cylinder, it is necessary to compute but one-half of it and hence only half is shown. That section of the frame to be taken is the dangerous section, that part at which the cross section of the frame is the smallest, and the metal is nearest the center or on the line *A-B*, Fig. 62, or *A-B*, Fig. 60.

The tensile stress is the total maximum pressure on the piston, and in pounds per square inch is the total stress divided by the area of the section. The maximum pressure on the average small or medium size gas engine can usually be taken as 350 lb. per sq. in. On very large engines with water-cooled pistons the maximum pressure may run as high as 450 lb. per sq. in.

On the engine in question, the maximum pressure is $\frac{20^2 \times .7854 \times 350}{2} = 54{,}980$ lb. on one-half frame. The area of the section is 77.24 sq. in., hence the tensile stress is $\frac{54{,}980}{77.24} = 711.8$ lb. per sq. in. The moment of the pressure is $54{,}980 \times 9 = 494{,}820$ inch pounds.

The fiber stress at the outer section is found by dividing the moment in inch-pounds by the modulus of section. The modulus of section (Z) is the moment of inertia (I) divided by the distance (z) of the outer fiber from the neutral axis.

To find the neutral axis, which is at the center of gravity of the figure, the simplest method is to lay out the section to scale on metal or stiff cardboard, cut it out and balance it on a sharp corner, as, for example, the corner of a triangular draftsman's scale. In the figure this is 9″ from the top of the frame. And as the height of the center of the cylinder is the same as that of the top of the frame, the moment arm a of the pressure is the same as z. If the center of the cylinder were above the top of the frame, this distance should be added to z to obtain the moment arm.

To obtain the moment of inertia of an intricate figure *like that in* Fig. 63, it is necessary to divide it up into a

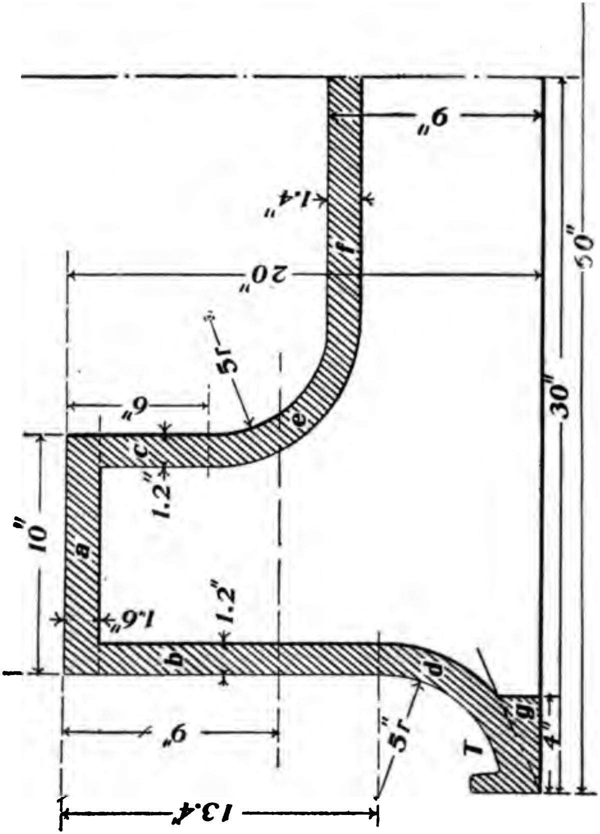

FIG. 63.

number of regular figures, and from the moment of inertia of these to compute the moment of inertia of the entire section. The figure shown may be divided into four parallelograms, two quarter rings and one triangle. The moment of inertia of each of these figures about the neutral axis of the section is the moment of inertia of the figure about its own neutral axis plus its area times the square of the distance of its neutral axis from the neutral axis of the section. Expressed as a formula, it is

$$I_s = I_f + A_f\, y^2$$

Wherein I_s = the moment of inertia referred to the axis of the section.
I_f the moment of inertia of the figure.
A_f = the area of the figure.
y = the distance from the axis of the figure to the axis of the section.

The moment of inertia of a parallelogram is

$$\frac{bh^3}{12}$$

Wherein b = the width,
h = the height.

As the area is equal to bh, the formula for I_s may be simplified for computation to

$$I_s = bh\left(\frac{h^2}{12} + y^2\right)$$

The triangle has $I = \frac{bh}{36}$

The quarter ring is more troublesome to handle, as its I about its c. g. is very complicated. It is simpler therefore to consider the moment about a line passing through the center of the circles of which it is formed. The quarter ring has $I = .1964(r_1^4 - r_2^4)$ about the center of the circles.

Wherein r_1 = radius outer circle.
r_2 = radius inner circle.

The I referred to the center of the section is

$$I_s = I_r + Ay^2 + 2\ Ayc$$

Wherein y = the distance of the base or center of the circles to the c. g. of the section.
c = the vertical distance from the c. g. of the ring to a horizontal line passing through the center of the circles.

This becomes

$$I = .1964\,(r_1^4 - r_2^4) - .7854\,(r_1^2 - r_2^2)\,y^2 - .6736\,(r_1^3 - r_2^3)\,y.$$

That the reader may understand the method more fully, the computation is given in full. The letters in parentheses correspond to those on the various sections in Fig. 63.

$$
\begin{array}{lll}
(a) & 10 \times 1.6\left(\dfrac{1.6^2}{12} + 8.2^2\right) & = 1079 \\
(b) & 1.2 \times 11.8\left(\dfrac{11.8^2}{12} + 1.5^2\right) & = 196.1 \\
(c) & 1.2 \times 4.4\left(\dfrac{4.4^2}{12} + 5.2^2\right) & = 150.4 \\
(d) & .1964\,(6.2^4 - 5^4) + .7854\,(6.2^2 - 5^2) \times & \\
 & \quad 4.4^2 + .6736\,(6.2^3 - 5^3)\,4.4 & = 707.7 \\
(e)^{*} & .1964\,(6.3^4 - 5^4) + .7854\,(6.3^2 - 5^2) \times & \\
 & \quad 3^2 + .6736\,(6.3^3 - 5^3)3 & = 546.6 \\
(f) & 10 \times 1.4\left(\dfrac{1.4^2}{12} + 2.7^2\right) & = 103.2 \\
(g) & \dfrac{4 \times 1.8^3}{36} + 4 \times .9 \times 10.4^2 & = 396 \\
\hline
 & \text{Total I for section} & = 3179
\end{array}
$$

$$\text{Section modulus } Z = \frac{3179}{9} = 353.2.$$

$$\text{Maximum fiber stress} = \frac{494{,}820}{353.2} = 1{,}400 \text{ lb. per sq. in.}$$

Total stress then becomes 711.8 + 1,400 = 2111.8 lb. per sq. in.

If the maximum pressure be taken at 450 lb. per sq. in., the total stress would be 2-7ths more, or 2,714.8 lb. per sq. in., which is close to the limit given above (2800 lb.)

If the center of the cylinder were to be taken at 4" above the top of the frame, for example, the moment arm would increase to 13" and the total stress to 2733.3.

By assuming the center of the cylinder to be placed 4½" below the top of the frame, the moment arm would be reduced to 4½", the fiber stress to 700 lb. and the total stress to 1,411.8 lb.

These two latter stresses are based on a maximum *pressure of* 350 lb. per sq. in. With the center of the *cylinder 4" above* the top of the frame, the total stress

---

[*] *taken as mean radius.*

on the basis of 450 lb. maximum would increase to 3,486.2 lb. per sq. in, and to meet this condition it would be necessary to increase the widths of the frame members by approximately 25%. This illustrates quite forcibly the advantage of keeping the top of the frame as high as or higher than the center of the cylinder.

For the side-crank engine there is the sidewise or horizontal bending moment to reckon with. The moment arm in this case is the distance from the vertical neutral axis of the frame to the center line of the cylinder measured in a horizontal direction. In this case the moment of inertia of the frame should be taken about its vertical neutral axis. The treatment is the same as for Fig. 63.

The designer should keep in mind convenience of access to the various parts for repairs and construction, and design the frame accordingly. The oil trough *T* around the lower edge of the frame is not absolutely necessary, but assists greatly in keeping the foundation and the floor of the engine room clean.

CHAPTER XXV.

FLYWHEELS.

A single-cylinder single-acting four-cycle engine requires a very heavy flywheel. This is because it not only has but one power stroke in four, but during one of the powerless strokes, a portion of the energy stored in the flywheel must be used to compress the charge. If, furthermore, the power impulse is made still more intermittent than one power stroke in four, as in hit-or-miss regulation, a much greater weight is required than for governing by reducing the force of the power impulse.

For different requirements in speed regulation there will be a difference in the weight of the flywheel. Steadiness of speed is usually spoken of in terms of percentage variation, for there is no such thing as absolute uniformity of speed, but there can be obtained a variation between limits that are determined by the requirements of the machinery to be driven and which depends upon the efficiency of the governing mechanism and the regulating power of the flywheel.

For example, an engine that is operating with a speed variation of 2% and at an average speed of 200 r.p.m., has a difference between the highest and lowest speed occurring between one power impulse and the next, of $200 \times .02 = 4$ r.p.m. Therefore, since the average speed of the engine should be 200 r.p.m., the maximum variation of speed should be between 198 and 202 r.p.m.

The mistake is quite frequently made of considering the steadiness of speed, due to the flywheel, as based upon the difference between the speed at no load and at full load. This is a matter that is dependent upon the efficiency of the governing mechanism.

The steadiness of speed between one impulse and the next is dependent entirely on the capacity for storing energy that is contained both in the flywheel of the engine *and in the moving* parts of the machinery which it is driving. Thus, the armature of a dynamo adds somewhat to the capacity of the flywheel, as do the pulleys and similar members of the driven machines.

In the operation of belt driven generators for electrical work, the capacity of the engine flywheel is quite frequently augmented by the use of an additional flywheel on the armature shaft, occasionally on a jack-shaft between the engine and the dynamo, and even on both jack-shaft and dynamo shaft.

Except in driving electric generators, no account is taken of the capacity of the driven machinery. When the large armature of a direct driven generator is on the engine shaft, it is quite frequently of sufficient weight to provide all the flywheel capacity necessary. In computing the weight of a flywheel for a jack-shaft or for the armature shaft of a belt-driven dynamo the designer should not overlook the higher speed of the armature, and the fact that the speed of the shaft on which the flywheel is attached is the speed to be employed in the computation.

The theoretically correct method of computing the required capacity of a gas engine flywheel is to lay out the tangential effort diagram based upon the indicator diagram to be expected from the engine and the effects of the inertia of the reciprocating parts. The discussion of these methods would require more space than is at the author's disposal in this work. However, the following formula for the weight of the flywheel rim will be found to meet average conditions. It is based on the consideration that in a flywheel of customary proportions one-tenth of the required capacity is taken care of by the spokes and the balance by the rim proper.

The basic formula applies to a single-cylinder single-acting four-cycle motor governed by the variable impulse system.

Let $W =$ the weight of the rim in pounds;

$f =$ the diameter at the center of gravity of the rim in inches;

$N =$ the r.p.m.;

$E =$ the coefficient of unsteadiness permissible;

$$\textit{Then } W = \frac{\text{I.H.P.} \times 111{,}600{,}000{,}000.}{f^2 N^3 E}$$

The values of *E* usually employed are given in the following table:

For pumping	.05
For ordinary machinery	.025
For textile machinery	.02
For spinning machinery	.01
For D. C. generators	.0125
For A. C. generators	.008
For A. C. generators in parallel	.006
For marine engines	.15
For automobile motors (1-cyl.)	.335*
Ditto—two cylinders	.550*
Ditto—four or more cylinders	.650*

For other than single-cylinder single-acting four-cycles the following ratios of *W* are given by Gueldner:

No. Cylinders	Arrangement	Ratio 4-cycle	Ratio 2-cycle
1	Hit-or-miss	2	
1	S. A.	1.	.401
1	D. A.	.615	.106
2	S. A. tandem	.398	.399
2	S. A. double opp.	.398	.399
2	S. A. cranks together, in line	.396	.401
2	S. A. cranks 180°	.645	.084
3	S. A. cranks 120°	.226	.0395
4	S. A. cranks 180°	.084	

Abbreviations: S. A., single acting; D. A., double acting.

To use the tables, compute the weight of the flywheel by means of the formula, employing the value of *E* designated in the first table. Multiply this result by the factor in the second table.

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* The flywheel of an automobile motor should be computed for 300 r.p.m. and for the power the motor would develop at that speed. The reason for using this low speed is that it is the average minimum speed of the motor when allowed to run idle in the car. The value of *E* for single-cylinder motors is smaller than given by some authorities, but it is the figure employed for the single-cylinder horizontal motors so popular in the early days of the automobile. While the values for two-and four-cylinder motors do not appear consistent with that for the single cylinder, they are, nevertheless, general practice.

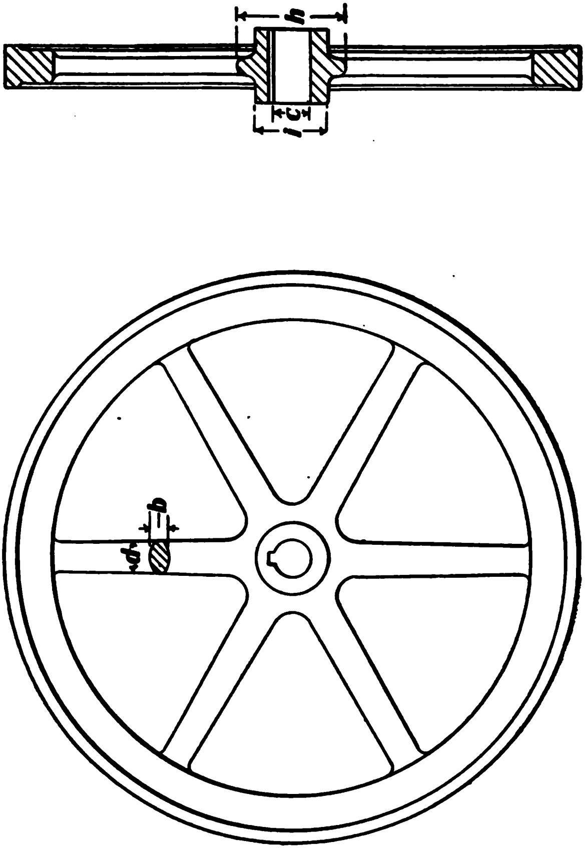

FIG. 64.

The general proportions of the average flywheel are shown in Fig. 64. The outside diameter of the wheel is generally made, for stationary engines, from four to five times the stroke, although in very large engines it will be found, at times, to be nearly seven times the stroke. For marine engines from 3½ to four times the stroke is usual and about 3.2 stroke for automobile engines. The proportion of the parts are based upon the diameter of the shaft as a unit and are as follows:

$i = 2c$;
$h = 3c$;
$l = 1.75c$ to $2.5c$, and not less than $1.5c$:
$d = .8c$ to $1.2c$;
$b = .4d$ to $.6d$ approximately.

The latter dimension should be computed from the following formula:

$$b = \frac{.316\ D^2L}{d^2n}$$

Wherein D = diameter of cylinder in inches;
L = stroke in inches;
n = the number of spokes.

It is the usual practice to use six spokes for wheels up to 10 feet in diameter, and above that eight are usually employed.

The proportions of the rim should be according to the requirements of the engine. If the flywheel is to be used as a pulley it should be made to correspond to the width of the belt. The wider the rim the greater will be the diameter of the wheel at the center of gravity and the lighter it may be made for a certain required capacity. The linear speed of the rim should not exceed 6,000 feet per minute, and the maximum diameter should therefore be not greater than that given by the following formula:

$$D = \frac{1910}{N}$$

Wherein D = the outer diameter of the wheel in feet;
N = r. p. m.

The general proportions of a flywheel designed for *small marine* engines are given in Fig. 64 *A*. Quite a wide *variation of proportions* is to be found in this type of *motor.*

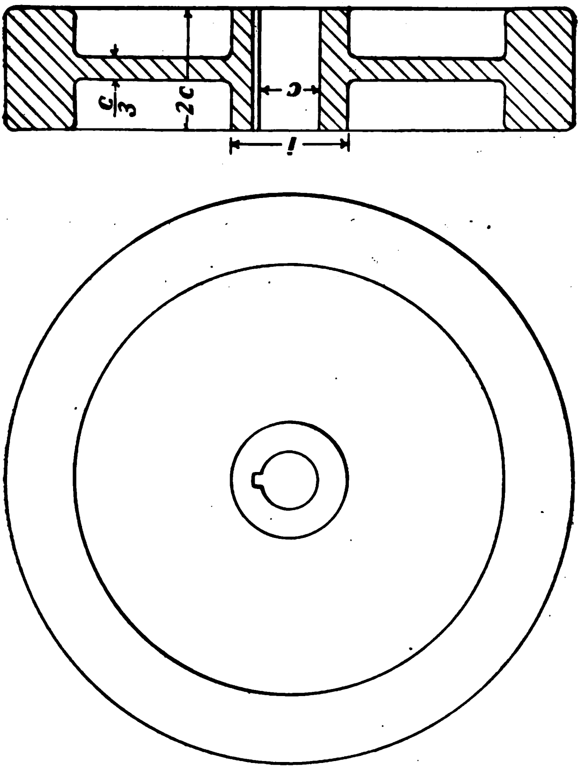

Fig. 64A.

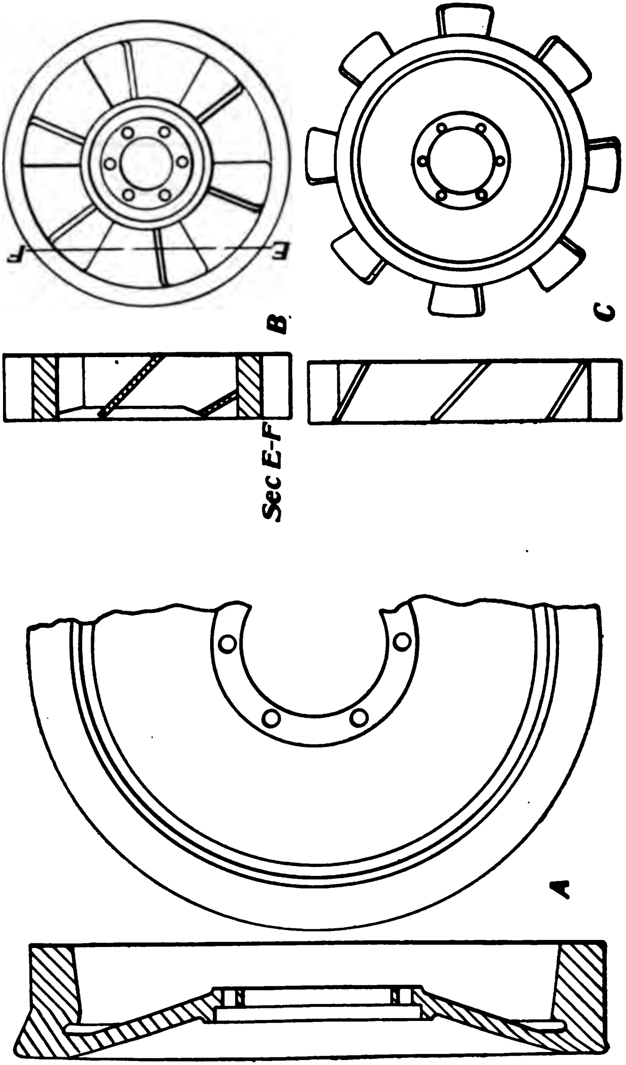

FIG. 65.

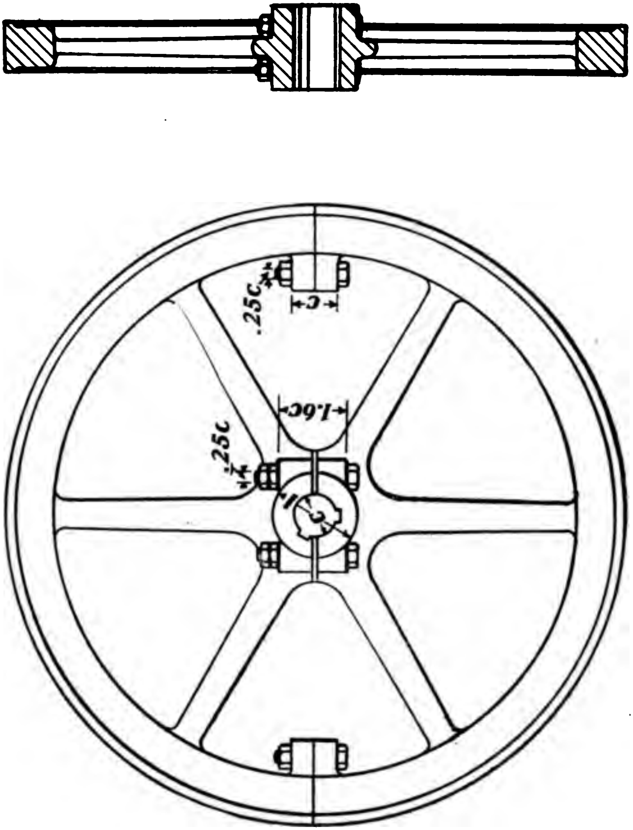

FIG. 65D.

Fig. 65 shows at *A* an example of the ordinary type of automobile engine flywheel tapered for a cone clutch. The manner of incorporating the fan in the flywheel of an automobile engine is shown at *B* in the figure. The fan blades when incorporated in the web take the place of spokes and are cast of the same material as the wheel. Outside blades are made of comparatively heavy sheet iron, the form being shown at *C*. These are laid in the mold and the casting is poured around them. A very effective joint is made in this way and much better than by attaching the blades in slots milled in the rim.

Fig. 65 *D* shows the proportions of a flywheel for large stationary engines in which the hub is clamped on the shaft. Wheels of this character are generally made in two pieces. If over 15″ wide, two rows of spokes are used.

CHAPTER XXVI.

GOVERNOR DESIGN.

Methods of regulation were fully discussed in Chapter XII, and the present chapter will be confined to the mechanism employed to bring these methods into play. As it is obviously impractical to illustrate every governor arrangement, the principal types will be discussed, and examples shown of some of the most widely used governors.

While the function of the flywheel is to steady the speed of the crank-shaft *between* impulses, that of the governor is to apportion the power to the resistance which is to be overcome, and without fluctuation of speed beyond a certain limit made necessary by the requirements of the machinery driven. The requirements of speed control for various duties have been given in Chapter XXV.

Any moving body requires a certain amount of force to keep it in motion at a certain speed. If the speed of the moving body is increased (accelerated), an increased force corresponding to the acceleration is required, and any decrease of the moving force results in decrease in the speed of the body (negative acceleration). It is this property of moving bodies that is made use of in all mechanism employed for speed control.

The force of gravity on a weight falling in a vacuum, so that the resistance due to the air is eliminated, causes it to fall 16.1 feet during the first second, with an acceleration of 32.2 feet per second for each second of fall. Therefore the force of gravity, usually denoted by the letter g, is 32.2. The mass of any body is the quotient of its weight divided by 32.2, and is expressed by the equation:

$$M = \frac{W}{g}$$

There are quite a number of mechanical devices employed for controlling the speed of a gas engine. These are based on the following principles: The tendency of a pendulum to swing for a constant period of time for one oscillation (swing from side to side), the resistance of a moving weight to acceleration (inertia), the force exerted by a revolving weight on the arm connecting it to the center of rotation, the force required for a plunger to drive air through an aperture (pneumatic governor), the force required to drive water through an aperture (hydraulic governor), and the force required to drive an electric generator (electric governor.)

The pendulum and the inertia governor are employed on engines governing by the hit-or-miss method, either by deflecting a part of the operating mechanism employed to open the gas valve or throwing a stop in the path of the exhaust valve to hold it open. The centrifugal governor is that in most general use for engines employing the variable impulse method, and is the most accurate of all types with the possible exception of the electric governor. In the latter, the principle employed is the increase in voltage caused by increase of speed in the electric generator or dynamo.

Two forms of pendulum governor are shown in Fig. 66. At *A* and *B* are shown two positions of a pendulum governor controlling the gas valve. On the push rod *p* actuated by a cam on the lay shaft is carried a swinging weight *w* adjustable along the arm *b*, of a bell crank. On the arm *a* of the bell crank is a knob *k* which comes in contact with a fixed knob *m* as the push rod moves to the left, and the bell crank is rotated in a clock-wise direction. So long as the speed of the engine is not too great, the weight will swing back and allow the point of the pick blade *c* to engage the notch on the end of the valve stem *g*, as shown in the sketch at *A*.

In case the speed of the engine rises above the normal, the pendulum has not time to swing to the right far enough for *c* to engage *g*, and it misses the notch. In consequence the gas valve is not opened and an impulse is missed. Lowering the weight on the arm *b*, makes the oscillation of the pendulum slower and reduces the speed of the engine. Raising the weight has the opposite effect.

In sketch *C* is shown one method of applying the pendulum governor to the control of the exhaust valve. *In case the* speed of the engine rises above the normal, *the weight w lags* behind when the push-rod moves to the *right. Arm a pushes* the catch *c* in the path of the collar

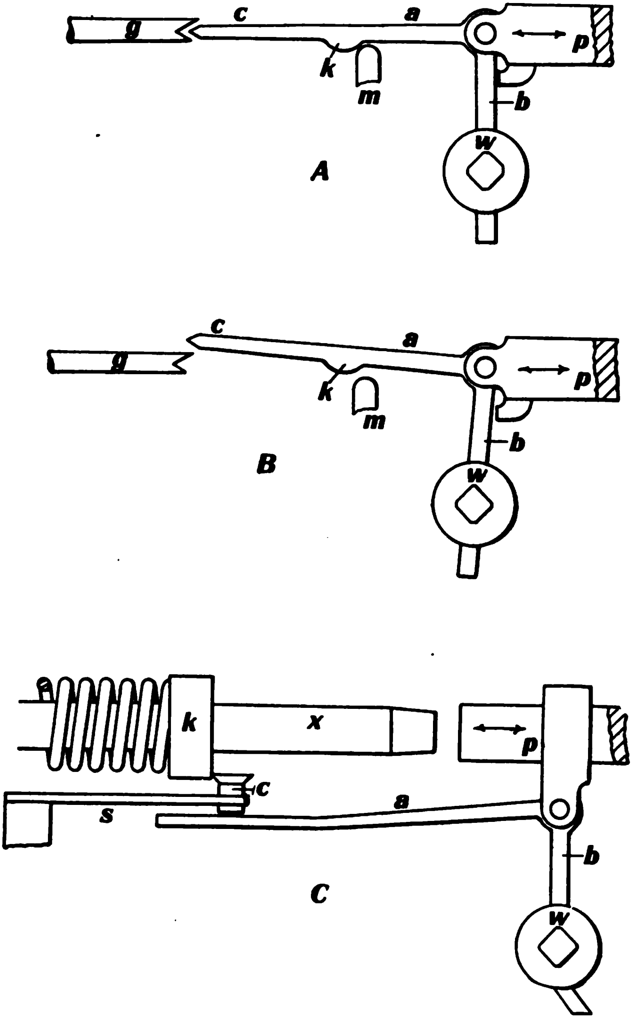

Fig. 66.

k, and holding the exhaust valve stem *x*, the valve is prevented from closing. On the next left hand movement of the push-rod it comes in contact with *x*, moving the valve sufficiently to allow the catch to release under the action of the flat spring *s*.

The theoretical principles involved in the time of swing for a pendulum are not realized in practice, and therefore a considerable range of adjustment of the weight should be provided for.

The principles of the design are based upon the following considerations, and by the use of these principles and means for adjusting the governor by trial, the desired results may be obtained.

For a mechanism such as that illustrated in Fig. 66 *A* and *B*, the stroke of the push-rod should be divided into three periods. The first period is that from the beginning of the stroke to the passage of the knob *k* over the knob *m*. The second period is from the passage of the knobs to the point designed for the meeting of the pick blade *c* and the valve stem *g*. The third period is that from the meeting of the blade and the valve stem to the end of the stroke.

It is with the time of the second period only that we are concerned in the computation of the governor. The actual time of this period should be that from the start of the free swinging of the pendulum until the point of *c* is opposite the lip of the notch on the valve stem. This time should be derived from the corresponding cam movement.

The next step in the computation is to find a pendulum which will swing through the required distance in the time of this second period. The swing usually employed for a pendulum governor is one-half of the single oscillation, or to the point at which the pendulum hangs vertically. The formula for the time of the half oscillation is

$$t = 1.571\sqrt{\frac{I}{Ws}}$$

Wherein t = the time in seconds;
W = the weight of the pendulum;
I = its moment of inertia;
s = the distance in inches from the pivot x to the center of gravity of the pendulum.

As the moment of inertia is that about the pivot, it is necessary, in order to secure an accurate result, to compute the moment in the same manner as when computing the fiber stress in the engine frame. However, a sufficiently close approximation may be obtained by using the moment of the weight alone, referred to the pivot.

If I_o is the moment of inertia of the weight about its center of gravity, the moment about the pivot will be

$$I = I_o + Ms^2.$$

The moments of inertia of the cylinder and the sphere, which are the forms most generally used are given below.

For the cylinder about a radius of the mid-circular cross section

$$I_o = M \left(\frac{r^2}{4} + \frac{h^2}{12}\right)$$

Wherein r = the radius of the circular base;
h = height between circular bases.

For the sphere about a radius

$$I_o = .4\ Mr^2$$

r = the radius of the sphere.

It will be noted that what is actually wanted is the determination of the value of s. While it is quite a simple matter for the mathematician to evolve such a formula, the formula itself is unwieldy and much better speed may be made by the use of the formula for time. The formula for s is as follows:

$$s = 2.467 \frac{I}{Wt^2}$$

The difficulty with this formula is that in substituting for I its value in terms of the moment of inertia of the weight about its center of gravity, s enters into the right hand side of the equation. For the interest of those who may wish to *compute* the value of s, the following gives the value direct.

$$s = \frac{t}{3.1416} + \frac{1}{6.2832t}\sqrt{\frac{97.47\ I_o + 4t^2W}{W}}$$

Formulas for the various centrifugal types of governors are fully discussed in all the more important works on steam engine design. For a complete discussion of the various forms of this type the reader is referred to *such* works. The following discussion will be confined to the general principles involved and their application to gas engine speed control.

The application of the centrifugal type of governor to hit-or-miss regulation, is illustrated by the example shown in Fig. 67. On the crank-shaft *C* is a fixed collar *k*, to which, at *pp*, are pivoted the levers *ll* carrying the balls *bb*. As the crank-shaft revolves, the balls tend to swing in the direction of the arrows, owing to centrifugal force, but are withheld by the springs *s*, only one of which appears in the figure. When the force acting on the balls is sufficient to overcome the tension of the springs, the balls fly outward, forcing the collar *c* toward *X*. This collar carries a ring running in a groove and to this ring is pivoted the lever *m* with a fulcrum at *p*. The long end of the lever controls the shaft of a rocker arm on which is the pick blade *x* designed to engage the stem *v* of the gas valve. As soon as the speed of the engine rises above normal, the governor forces the pick blade out of line with the valve stem, and the engine misses its charge of gas.

Fig. 68 is a diagrammatic illustration of the application of the fly-ball type of governor to a gas engine throttle. The balls *bb* tend to swing by centrifugal force in the direction of the arrows, being restrained by the spring *s* bearing against the collar *c*. As the balls move outward they carry the collar downward and with it the stem *S* of the throttle valve, which is so guided that it does not rotate. Gas is drawn into the space in the center of the valve *V* by the suction of the engine from the gas passage *G*, and air from the passage *A*. As the valve is moved downward by the action of the governor, it gradually closes the narrow openings in the wall of the valve, and the entrance of the gas and air is throttled. The governor is rotated by means of the bevel gears *q* and *p*, of which the latter is driven from the engine by means of a belt. General practice on modern designs is to use a gear driven governor and dispense altogether with the belt.

The centrifugal force acting upon the balls or other weights of the centrifugal type of governor increases as the square of the speed and directly as the radius, according to the formula

$$F = Ma^2r$$

Wherein F = centrifugal force;

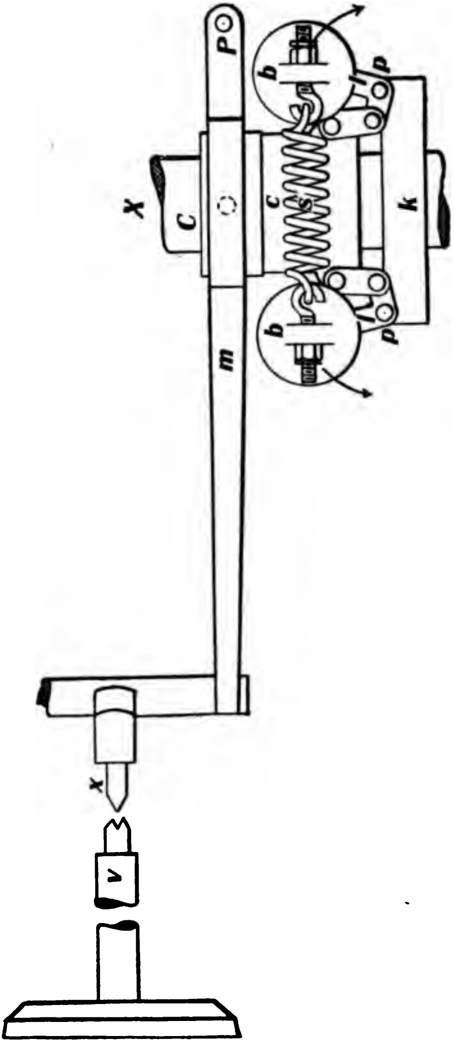

FIG. 67.

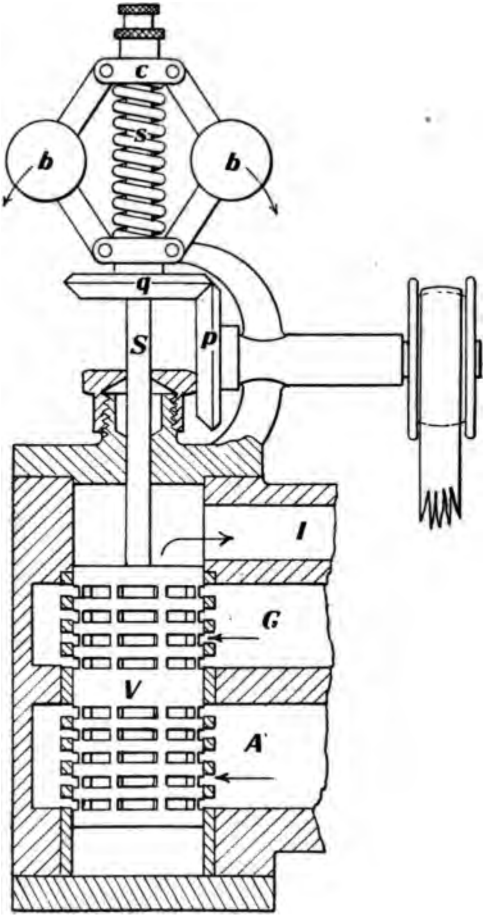

FIG. 68.

M = mass of weights.
a = angular velocity = $6.2832\ N$;
N = r. p. m. ÷ 60;
r = radius in feet of the center of gravity of the weights.

The great objection to the earlier types of fly-ball governor was that as the load on the engine decreased, the speed of the engine must increase a slight amount in order to move the regulating portion of the governor, and for each position of the governor the engine must run at a different speed, faster than the normal for light loads and below the normal for heavy loads. By examining the formula it will be noted that for a certain weight of the moving part of the governor the mass would remain constant, therefore when F varies either a or r or both must vary. If the governor is so designed that the speed of the engine is the same for all positions of the regulating mechanism, both M and a are constant and r varies with the centrifugal force. Any governor which meets this condition is said to be isochronous, and it will run at the same speed no matter what the length of the radius r.

One of the best examples of an isochronous governor is that shown in Fig. 69. In the figure the weights w move outward against the restraint of the springs s. As the weights move outward the levers shown draw the collar c upward with a corresponding movement of the throttle or other governing mechanism. It will be seen that the compression of the springs is exactly equal to the movement of the weights. Therefore to secure isochronism the scale of the spring should be equal to the value of Ma^2, when a is the angular velocity at which the governor is desired to run.

It should be noted that, in action, there must be a slight increase in speed in order that weights move outward, and a decrease of speed to cause the weights to move inward. So soon, however, as the weights move in either direction the governor tends to produce a condition of equilibrium between the spring and the force of the weight. As this condition is secured at one speed only, the engine will run at one speed irrespective of the load

A strictly isochronous condition of the governor is *not always* desirable. For example, on a single cylinder *four-cycle*, even if double acting, the impelling force on

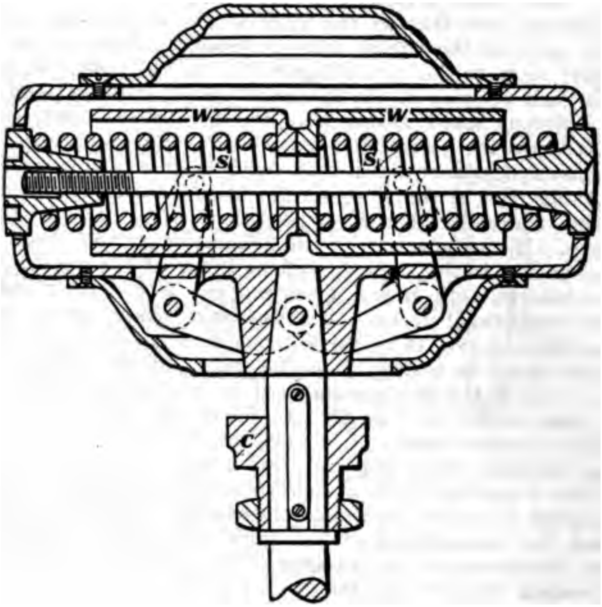

FIG. 69.

the piston is so variable that the position of the governor may change during the cycle and bring about that disagreeable action known as *hunting*. This action may to a certain extent be modified by the use of a dash pot in connection with the controlling mechanism, and the use of the dash pot is quite frequent in connection with a gas engine governor, especially with shaft governors. A small amount of friction will sometimes assist in steadying the action of a governor. The range of speed required for a gas engine governor to move the mechanism from the no-load to the full-load position should always be slightly greater than the coefficient of unsteadiness of the flywheel to avoid hunting due to the change of the pressure on the piston during the cycle. Therefore as the number of impulses per revolution increases the greater may be the sensitiveness of the governor.

Shaft governors, so frequently employed in steam engine practice to govern the engine by changing the throw of the eccentric, are used to but a small extent for gas engines, and therefore a discussion of their properties will not be given. On account of their great simplicity and their accuracy of speed control, they are deserving of attention, and they are employed with considerable success on several American designs.

CHAPTER XXVII.

BALANCING.

The balancing with a revolving weight of the moving parts of any engine in which all of the force is applied to one crank-pin, or of engines with more than one crank-pin in which the cranks are not at 180°, is a compromise only. For example, were we to balance all of the moving parts, both the reciprocating and the rotating, of a horizontal engine, it would be out of balance vertically. Were we to balance only the rotating parts the engine would be out of balance horizontally. The compromise usually employed is to balance all of the rotating weight and one-half of the reciprocating weight.

The rotating weight consists of the crank arm, the crank-pin and a portion of the connecting rod. The reciprocating weight includes that portion of the connecting rod not considered as rotating, the piston, the cross-head and any parts attached to these, such as tail rods or other appurtenances.

That portion of the rod to be considered as rotating may be found as follows: Support the piston end of the rod on a knife edge opposite the center of the bearing in a horizontal position with the other end supported in a similar manner on a scale. The weight shown by the scale should be considered as rotating. If, however, five eighths of the rod be considered as rotating the result will be very close.

The formula for the balance weight will be as follows:

Let B = weight of both balance weights;
M = weight of crank-pin + rotating part of rod;
K = weight of the reciprocating parts, including remainder of rod;
J = the weight of both crank arms;
m = crank radius = one-half the stroke;
j = the distance from the center of gravity of the crank arm to the center of the shaft;
q = the distance of the center of gravity of the

balance weight from the center of the shaft.

$$\text{Then } B = \frac{\left(M+\frac{K}{2}\right) m + Jj}{q}$$

The best place for the balance weight is on the crank arms of the engine, and it is attached on small engines by riveting and on large engines by bolting. In designing the fastening of a balance weight, the centrifugal force of the weight should be figured for the highest probable speed of the engine. This is especially important in automobile or marine engines, the operators of which often allow them to race to speeds of 1,500 to 2,000 r. p. m.

On small and cheap engines the balance weight is quite often made a part of the flywheel. While the weight of the balance may be reduced in this way on account of the larger radius available, it is hard on the shaft and is inclined to bend the shaft, unless made especially heavy to withstand the bending action. On no account should a balance weight be placed in the flywheel when but one flywheel is used.

The balance of multiple cylinder engines depends upon the arrangement of the cylinders, and the type of the engine whether two or four-cycle. In a two-cylinder engine with cranks at 180°, the parts assume very nearly opposite positions at all points of the stroke, with the exception that there is a tendency toward a slight rocking motion in the case of a two-cycle, on account of the force on one piston being slightly greater than on the other. In the four-cycle, the uneven division of the impulses with the cranks at 180°, throws the engine out of balance. For this reason, two-cylinder four-cycles with the cylinders parallel are arranged with the cranks at the same side of the shaft and the balance is effected with weights.

The three-cylinder two-cycle with the cranks at 120° is very closely in balance, but the four-cycle three-cylinder is so uneven in the sequence of its impulses, that it will vibrate excessively, especially at high speeds and light loads.

The *four*-cylinder is, however, in best balance in the *four*-cycle type. For example, in the vertical engine the two end cranks are set to go down as the two middle cranks go up, and with the crank-pins all in one plane, see Fig. 70 *A*, the impulses may be evenly spaced as to time. In the four-cylinder two-cycle there are four impulses in each revolution and therefore the cranks must be arranged 90° apart. The arrangement usually used by the author is shown in Fig. 70 *B*. The crank-pin *a* and *b* are at 180° to each other, as are *c* and *d*. Cranks *b* and *c* are placed at 90°, and with the crank-shaft turning in the direction of the arrow the order of firing is *a-d-b-c*. Ordinarily, this arrangement gives a fairly steady engine, unless it is fitted with a weak foundation, as in some types of aeroplanes.

In the six-cylinder motor, of either the four- or the two-cycle type, the most perfect balance is obtained. The best arrangement for a four-cycle six, is that shown at Fig 70 *C* in which the two center cranks are rotating together, those on each side of the two center cranks follow at 120°, and those in turn are followed by the two end cranks at 120°.

In the six-cylinder two-cycle the arrangement at Fig. 70 *D* gives good results. Cranks *a, b* and *c* are arranged at 120° to each other, and cranks *d, e* and *f* likewise while cranks *c* and *d* are at 60°. This gives an excellent mechanical balance and, as the six impulses are evenly divided through a single revolution, the operation of a six-cylinder two-cycle is very smooth and with no vibration.

It is most important in the construction of multiple cylinder engines that all the pistons, and the rods be of the same weight, or at least that the maximum variation in weight does not exceed one per cent. Thus for pistons weighing 4 lb. 2 oz. the difference between the weight of the heaviest and the weight of the lightest piston should not exceed ½ oz. The most accurate way to balance the connecting rods is to swing them from the center of the piston pin bearings as pendulums, and so balance them that the time of swing is the same for each. Practically as close results may be obtained by weighing the heavy ends and the light ends separately and making the rods equal in this respect. For drop forgings or shapes in which there is likely to be little difference in

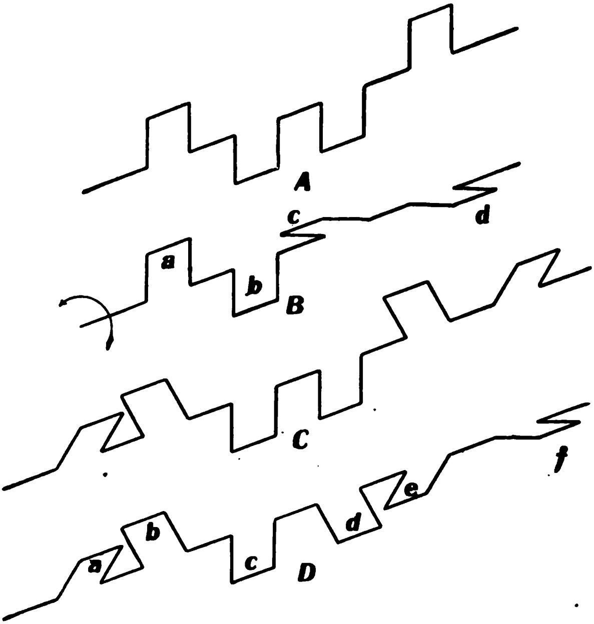

FIG. 70.

the weights of the two ends, weighing the entire rod will suffice.

It is hardly necessary to remind the reader that the flywheel should be carefully balanced. Usually a flywheel is considered sufficiently well balanced when it is in static balance. In other words, when it may be placed on a mandrel and the mandrel on a pair of level straight-edges at any position without rolling. Later refinements call for a "running balance," which must be determined by means of a machine especially designed for this purpose. This is necessary only at high rotative speeds and is due to the fact that a wheel revolving at high speeds has a tendency to revolve about its center of gravity.

In using the balancing machine, the wheel, already given a static balance, is mounted on a pivot with the face of the wheel in a horizontal position. The machine is so arranged that the wheel is first rotated by the machine and then raised clear of the driving members and allowed to spin by its own momentum. Any tendency of the rim to run out of true is determined by holding a piece of wet chalk against it, and the unbalanced condition is corrected by means of weights. The static balance is not disturbed because the weights are moved along the rim in a direction parallel to the center of the shaft.

It is of importance that the crank-shaft of a multiple cylinder engine be balanced as is the flywheel, by rolling it on a pair of straight-edges to test it and removing metal from the heavy sides. After it has thus been placed in static balance, it should be placed in a balancing machine and tested for running balance.

Since the advent of extremely high speed for automobile engines, careful balancing of the reciprocating and the rotating parts, in a thorough manner, has become imperative. An unbalanced engine reaches its maximum power at 2,200 to 2,400 r. p. m. A well-balanced one will show increase of horsepower up to 4,000 r. p. m. and over.

CHAPTER XXVIII.

FOUNDATIONS.

If the foundation of an engine is inadequate, it may be expected that the engine will give more or less trouble from vibration, and in time work itself loose. Except in the case of very small gas engines, an engine should not be bolted to the floor for anything other than a temporary installation. On account of the nature of the operation of a gas engine it is more severe upon a foundation than a steam engine and a gas engine foundation should be of better grade than one for a steam engine. Even when an engine is installed upon the floor of a building it should be provided with a foundation swung below the floor in a hanging frame.

The material from which a foundation is built, is invariably some form of masonry, either concrete, stone or brick. On the top of the foundation may be set a capstone of granite, limestone, bluestone or any stone of close-grained structure. The ordinary sandstones are not firm enough for this purpose. The substitution of two parallel heavy timbers of oak or yellow pine has been successfully employed in lieu of a capstone and has the advantage of absorbing vibration when the engine is at all deficient in balance.

Concrete should be made of good sharp sand, a good quality of cement and broken stone. The stone should be of a size that will pass through a two-inch ring, but not through a one-inch ring. Gravel, broken brick or cinders have been used, the latter making a better concrete than might be supposed. For concrete the following mixture is recommended:

Cement	1 part;
Sand	3 parts;
Gravel or stone	5 parts.

The concrete should be laid in a crib in layers not over six inches deep, and each layer should be thoroughly tamped before the next is put in.

A very good foundation may be quickly and cheaply prepared in the following manner. Lay a brick wall in cement mortar as the outside boundary of the foundation. Then fill in the center with layers of brick or brickbats laid flat as in a wall and loosely. The space between the bricks should then be filled in with a mixture of one part cement to one or two parts of sand according to the quality of the cement. In case appearance is not a factor, that portion of the foundation below the surface of the ground may be laid without the trouble of making a wall. In this case, the earth surrounding the foundation should be sufficiently hard to serve as a temporary retaining wall for the brickwork.

All foundations should be allowed at least three days to set before the engine is put in place, and a week is a still better allowance. The foundation should be carried to solid rock or hard pan wherever possible, and it should, at the very least, be started from the ground. If necessary to hang a foundation from the upper floor, it should be placed as near to the wall as possible. In case the soil is loose and there is no hard pan or rock available, it may be necessary to drive piles to form the base of the foundation for a heavy engine.

When starting a foundation a wooden frame or templet should be made, and in this holes are bored to correspond in position to the holes in the frame for the foundation bolts. After the first two or three layers of the foundation are laid, the templet is lined up in the position chosen for the engine with the foundation bolts in place and the nuts on the upper ends of the bolts. The height of the top of the templet should be the same as the top of the bolt holes in the frame of the engine, when the engine is in place. The plates or nuts into which the lower end of the bolt is screwed should be in place, and, if any adjustment is required afterward, the bolts should be surrounded with pieces of gas pipe. When the engine is set the pipe may be filled with cement. A very simple way to hold the lower end of the bolts is to forge an eye on the lower end and pass through the eyes of each pair of bolts an iron rod. This effectually holds them down and keeps them from turning. Timbers instead of capstones are usually employed for small engines.

An investigation made by the author, the results of which were published in the September, 1916, issue of The Gas Engine Magazine, showed the following to be representative of American foundation practice.

The approximate weight of the foundation for engines of any type may be derived from the following formula:

Let F = the weight of the foundation in pounds;
E = the weight of the engine in pounds;
R = the r. p. m.;
K = a weight constant;
Then $F = K\ E\ \sqrt{R}$.

To secure the result directly in cubic feet of concrete, and using 133 pounds as the average weight of concrete per cubic foot, we have

$$f = k\ E\sqrt{R}.$$

in which
f = volume of foundation in cubic feet;
k = a volume constant.

A result of the investigation above referred to gives the following values for K and k for engines of different types:

Type of Engine.	K	k
4-Cylinder Vertical Gas Engine	.130	.000975
3-Cylinder Vertical Gas Engine	.150	.00113
2-Cylinder Vertical Gas Engine	.175	.00131
4-Cylinder Vertical Diesel	.177	.00133
Single Crank Double-acting Tandem	.320	.0024
Twin Crank Double-Acting Tandem	.19	.00143
Single Cylinder Horizontal Semi-Diesel	.3	.00225
2-Cylinder Horizontal Semi-Diesel	.24	.0018
3-Cylinder Horizontal Semi-Diesel	.23	.00173
4-Cylinder Horizontal Semi-Diesel	.225	.00069
Two-cycle Horizontal Semi-Diesel	.23	00173

The weight of brickwork is 112 lb. per cu. ft. in the wall or in the foundation. The average weight of concrete is 133 lbs. per cu. ft. It is not customary to make any difference in the design of the foundation to correspond to the difference in the weight of the two materials, although the weight of concrete is one-fifth more than that of brickwork. If but one foundation drawing is prepared, it is suggested that it be based on an average of the two-weights or 125 lbs. per cu. ft.

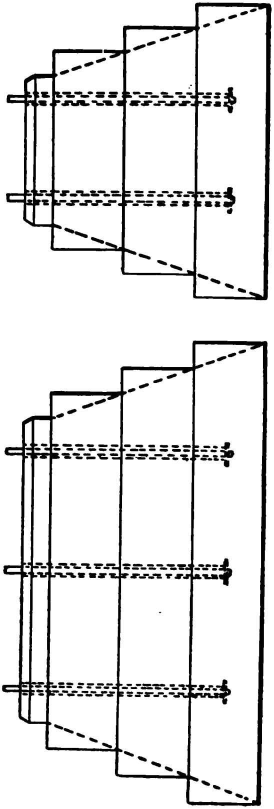

FIG. 71.

The bearing of a foundation upon the soil, where it can not be carried to bed rock, should be such that the pressure per sq. in. should not exceed the following values:*

Soft clay and fine damp sand	15 lb.
Loam, medium hard clay, dry sand with clay	30 lb.
Hard clay and dry sand	45-60 lb.
Hard bedded sand and gravel	75-90 lb.
First-class average soil	60-75 lb.
Ordinary average soil	45 lb.

An illustration of a common form of foundation is given in Fig. 71. The inclination of the wall from the top to the bottom is usually known as the *batter*, and it is customary to make it from 3" to 4" to the foot of height. Instead of building the foundation in tiers as shown, some designers prefer to make the slides with a gradual slope, as indicated by the dotted lines.

The number of foundation bolts is never less than four, and unless the engine is very small, six is the customary number up to about 50 H. P. While in engines of from 75 H. P. up we find them ranging from eight to as many as fourteen. The following formula will give the designer a close approximation to the size of the bolts:

Let H = the horsepower of the engine;
y = the area of one bolt at the bottom of the thread;
k = the number of bolts;

Then $$y = \frac{.115\,H + 1.35}{k}$$

While this formula gives a value to y that is very close to the average practice, the designer should check it by computing the strains on the individual bolts. The formula gives a size that will be generally found ample for engines of average design. See further notes on foundations in Chap. XXXII.

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*Gueldner, "Internal Combustion Engines."

CHAPTER XXIX.

MUFFLERS.

Mufflers are used on all engines of the internal combustion type with few exceptions. On the smaller sizes only the exhaust is muffled. On the larger horsepowers the noise produced by the suction of the charge is objectionable, and mufflers are used for the intakes of large engines as well as for the exhaust. Only on the motors used for racing automobiles, racing boats and aeroplanes do we find the muffler dispensed with. On the latter there is arising a demand for exhaust mufflers, especially where the aeroplane is to be used by the military for scouting purposes.

The muffling of the intake on stationary engines of moderate horsepowers is usually accomplished satisfactorily by drawing the air from a compartment in the base or the sub-base of the engine. Where small stationary gasoline engines are used in residences for power purposes as for suburban electric plants, a muffler should be applied to the carbureter intake. This is generally made of a short length of pipe or tube, stopped at the end and perforated on the sides. In order to avoid choking the intake, the combined area of the holes should be one-half greater than that of the intake. On engines of very large powers running into the hundreds, the intake is drawn through passages under ground lined with masonry. One of the important features of inlet passages of any kind is that they be free of dust.

For the exhaust, designs of mufflers vary greatly with the type of engine, and to some extent with the fancy of the designer. The principles involved are the emission of the exhaust gases from the final outlet to the atmosphere at the lowest possible velocity and the absence of resonance in the muffler itself. The reduction of the emission velocity may be accomplished in various ways. The gases may be expanded in an enlargement of the exhaust pipe, they may be passed back and forth over numerous obstructions or compelled to pass through a number of small holes, and they may be condensed by cooling, either by passing water into the exhaust pipe in direct contact with the gases or by water-jacketing the muffler. A further method is to enlarge the mouth of the exit pipe gradually by making it conical in form like the bell of a trumpet. The latter, while not used to any great extent, is very effective, and has the advantage that it materially reduces the back pressure on the engine.

Resonance of mufflers is avoided by covering them with some deadening material, such as asbestos or mineral wool, or by burying the muffler in the ground. The passage of water through the muffler also assists in over coming the metallic sound.

Mufflers should have a volume of at least four times the piston displacement of one cylinder, and from six to eight is a more effective ratio. For large stationary engines the best results are obtained by using separate mufflers for each cylinder.

Back pressure in a muffler usually reduces the power of the engine from 3 to 10 per cent., as compared with the power obtained from an open exhaust. On rare occasions a muffler has been known to add to the power of an engine. The author at one time built an experimental muffler comprised of a number of revolving fans rotated by the engine, which increased the power of the engine 3%. The muffling efficiency of the device was much better than any commercial form of muffler. It was not, however, a device that was practical commercially.

Several types of small mufflers which have given good results in practice are shown in Fig. 72. Muffler *A*, is an example of the concentric tube type. The gas passes into the outer compartment first, and is cooled and condensed if the muffler is not lagged. Generally the holes leading from the first compartment to the second are at the opposite end from the entrance. By placing them at the end nearest the entrance the current of gas reacts upon itself, increasing the muffling effect. The gases then pass through the various compartments, as shown by the arrows, the last compartments gradually decreasing in area to increase the period of emission to the atmosphere.

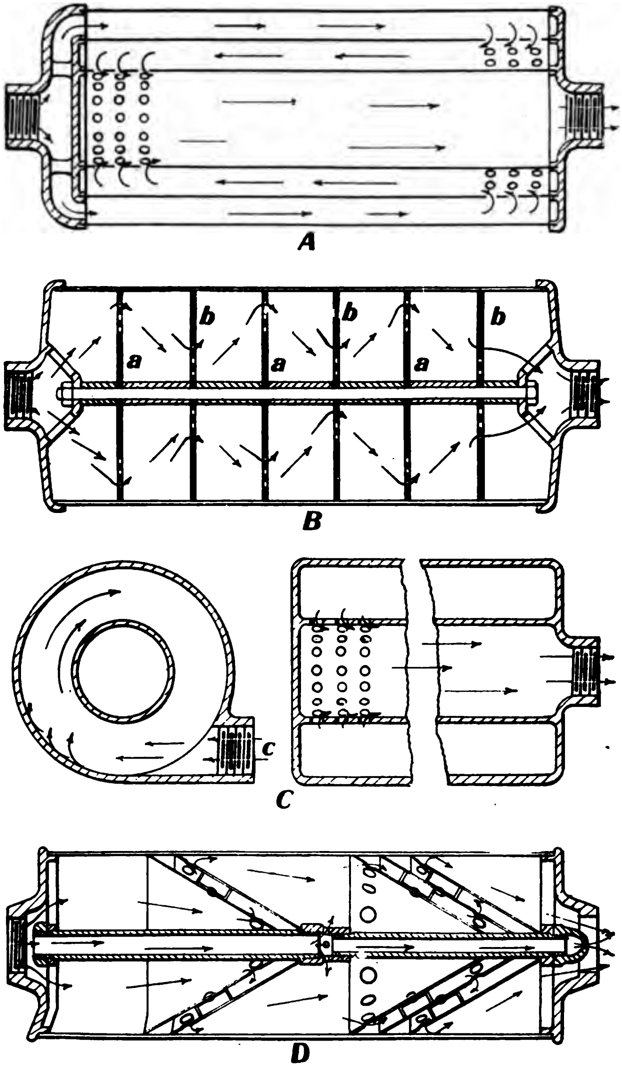


FIG. 72.

A very cheap and effective muffler is the baffle-plate form shown in Fig. 72 *B*. The punched plates *a* and *b* are strung alternately on the central rod, and the gases pass back and forth alternately from the center to the edge of the plates.

A muffler of the type shown in Fig. 72 *C* admits the gas at *c* against the outer wall, setting it in a swirl. It may be allowed to pass to a central compartment through the openings as shown and out at either one or both ends. Good results have been obtained by admitting the gas in the manner shown at one end of the chamber with emission at the other end, dispensing with the central tube.

The ejector muffler shown in Fig. 72 *D*, is a well known type of patented muffler. A portion of the gas enters the center tube while the balance passes through the baffle plates as shown by the arrows. A portion of the central stream leaves the tube as shown, while the balance passes out in the center of the exit through a small nozzle. The velocity of the gas through the tube is greater than that passing through the baffle plates and therefore a vacuum is formed both in the central chamber and at the exit on the principle of the steam ejector.

One of the faults of a great many mufflers is that they are not strong enough to withstand the strain of a muffler explosion. A muffler explosion that bursts the muffler is at least annoying, if not dangerous. This possibility can be avoided by designing the muffler to withstand a pressure of 100 lb. per square inch.

A muffler should be so designed that it can be readily taken apart for cleaning, as it is sure to become clogged with soot in time.

For those who are annoyed with the exhaust from large stationary engines, the author recommends the exhaust pit shown in Fig. 72 *E*. In the pit, lined with masonry, are placed a number of loose stones such as "niggerheads." The exhaust cooling water enters at *w* and is spread by the funnel *f*. The exhaust from the engine enters by means of the pipe *o* and is led to the bottom of the pit, whence it passes upward through the stones to the exit *e*. The mouth of the exit is bell-shaped as shown. While this form of muffling device may be

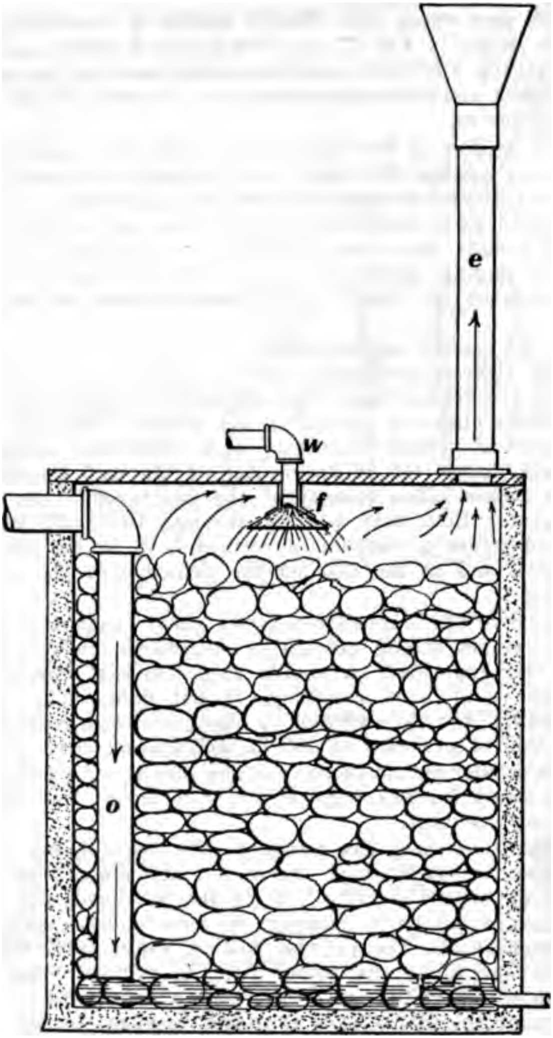


FIG. 72E.

costly as compared to the ordinary iron exhaust pot, it is warranted where the factory is located in a populous district, and the noise and vibration of the exhaust causes complaints from those residing in the neighborhood.

Under no circumstances should the exhaust from a gas engine be led into a chimney or into a sewer. There is always danger of a disastrous explosion from the accumulation of unburned gas due to missed explosions.

In boats, the muffler is frequently substituted by the underwater exhaust, a device of funnel shape so arranged on the bottom of the boat that rush of the water draws the gases with it. A long exhaust pipe, carried out through the stern, and through which the exhaust water is allowed to flow, will frequently cut down the noise effectively. This plan has the advantage that, if for any reason the cooling water ceases to flow, the noise of the exhaust increases and gives the operator warning.

CHAPTER XXX.

TWO-CYCLE ENGINES.

The author believes that nothing in gas engineering has been surrounded with more seeming mystery nor subject to more false notions than the two-cycle. This has been especially true of the smaller two-cycles with precompression of the charge in the base. For example, in the design of the ports and their timing, it is generally supposed that every new proportion of bore and stroke, must be the subject of endless experimenting before the correct sizes are secured. Another prevalent idea is that the bore of a two-cycle should be at least equal to, if not greater than the stroke. A notion also prevails that it is necessary to secure the highest possible compression in the base. Much time and money has been wasted in the mistaken idea that the introduction of the vapor of gasoline in the crankcase, cuts the oil and interferes with the lubrication. I have also heard the statement that the two-cycle would not pull (fall off in torque) at low speed, and again that no power could be secured above a certain speed, say 800 r. p. m.

Now, as a matter of fact the rules for the design of a successful two-cycle motor are few in number and easily understood. The peculiar ideas about this type outlined above, either have no basis in fact, or they are based on poorly designed engines. There is nothing very difficult about port design as will be shown. Two-cycles of all proportions of bores and strokes may be made successfully. If the designer wishes to follow any particular proportion of this kind it should be the proportion of the area of the piston to the stroke.

Crankcase compressions between 4½ and 10 lb. give good results within the usual range of speeds, and except, perhaps, for extremely high speeds, there is no advantage in going higher. The idea that the admission of gasoline to the base interferes with the lubrication, is laughable *in view of the fact* that the most successful way to lubricate the interior of a two-cycle is to mix the oil with the gasoline. A two-cycle of even *fairly* good design will hold to its load at low speed much better than a four-cycle designed for high speed. With the same careful attention to the design of the inlet and the exhaust passages that would be given to a high-speed four-cycle, equally good results may be obtained with a high-speed two-cycle.

In the layout of the port openings of the base-compression two-cycle, the angle of the crank-circle should be employed as a basis, just as in the four-cycle (see Chap. XXII). As the two-port type is designed upon the same principles as the three-port, the latter will be discussed first.

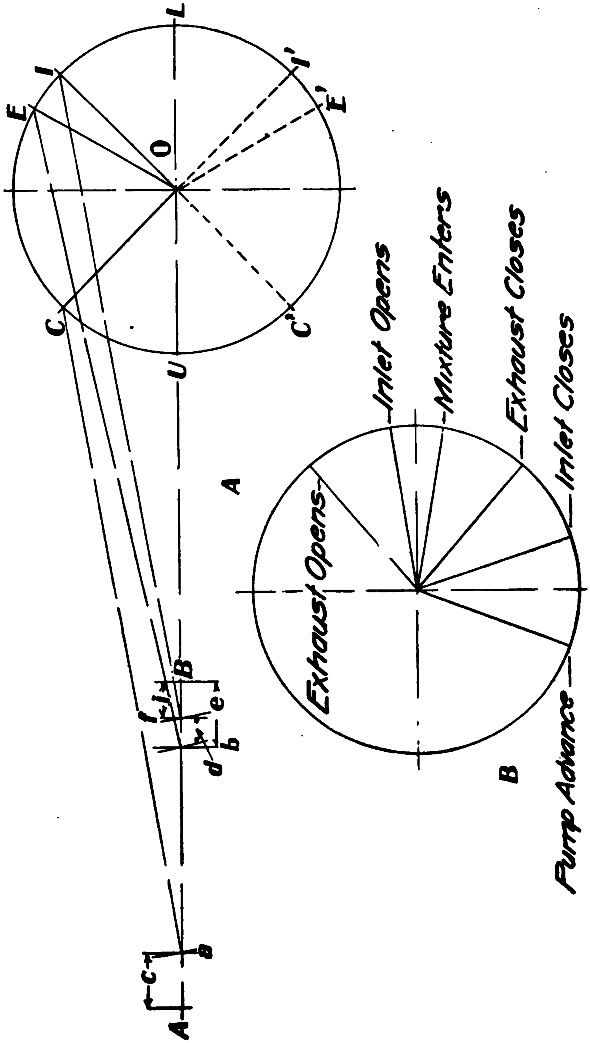
To begin with, the most important port in the design of a two-cycle is that leading to the crankcase. The first consideration in the design of any port is the crank angle at which it opens. In the use of one carbureter for all cylinders of a multiple cylinder engine, the ports should be so designed that no two are open to the carbureter at the same instant. In a four-cylinder with one carbureter, this necessitates that each port must be open less than 90° and 88° has shown satisfactory results. Unless the crankshaft is offset, the port will open at the same angle before the dead point as it will close after crankpin has passed the dead point. The opening will then be 44° ahead of the center. In engines of less than four cylinders or in which less than four cylinders are supplied from one carbureter, 48° has been used successfully. In automobile engines in which power at high speed is not so great a factor as good control at low speed, better results are obtained by opening 37° in advance of the center.

The area of inlet to the base should be estimated on the basis of the *effective* port area. The effective port area, is that area which gives the required speed of the gases based on the actual time during which the port is open. By referring to the valve diagram in Chap. XXII, it will be seen that inlet valve of a four-cycle may remain open as long as 210° of crank travel. In four-cycle practice the formulas are based on 180° duration of opening of the inlet. Assuming that the inlet port of a two-cycle is open 90° of crank-pin travel, the speed of the gases for the same area of inlet passages would be just twice that in a four-cycle at the same piston speed. Therefore, the effective area of the same port in the two-cycle would be but half that of a four-cycle. This particular point is one that is frequently overlooked in the design of a two-cycle, and is one of the reasons for lack of power at high speed in many designs. While it is desirable to limit the speed of the gases entering the engine to 100 feet per second, the author finds that it is usually quite difficult to get below 120 feet per second. Quite a number of existing designs show speeds as high as 190 feet per second.

Good results are usually obtained by opening the inlet port from the by-pass to the cylinder at 45°. As for the inlet from the by-pass to the cylinder, the gas has pressure in the base to drive it, the port may be about three-quarters the area of the inlet port to the base from the carbureter. The area of the cross-section of the by-pass should be no less than that of the cylinder inlet and may be made slightly larger to advantage. It is a mistake, however, to make the area of the by-pass too great, as too large a by-pass increases the volume of the base too much and lowers the compression in the base to a point below that required for good results.

The angle of the exhaust opening may be from 56° to 62° in advance of the dead point, depending to some extent upon the speed of the engine. Where possible, the area of the exhaust ports should be such that the relative speed of the gases does not exceed 80 feet per sec. As explained in Chap. XXII, this speed is an assumption only, and is based on what the speed of the gases would be if the contents of the cylinder were at atmospheric pressure and being driven through the port by the movement of the piston.

The port diagram of the average two-cycle with base compression is shown in Fig. 73 *A*. The crank circle is shown with its center at *O*, and the line *A-L* represents the line of piston travel. The angle *UOC* is the angle of cut-off for the crankcase port, also the angle of opening at *UOC'*. With the center at *C* and with a radius *Ca* equal to the length of the connecting rod, the point *a* is determined, also the width *c* of the crankcase inlet port. In *other* words, *c* is the measure of the piston travel from the *upper dead point* (vertical engine) to the point at which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Inlet Opens
Mixture Enters
Exhaust Closes
Inlet Closes
Exhaust Opens
Pump Advance
A
B

FIG. 73.

the piston closes the port. As it is customary to make the upper edge of the port even with the lower edge of the piston when the latter is at the top of its stroke, the distance of piston travel and the width of the port is the same.

Angle *LOE* is that of the exhaust opening with the corresponding angle of exhaust closing at *LOE'*. Point *b* indicates the position of the piston at the time of either closing or opening and *e* the width of the exhaust port, as it is customary to make the lower edges of both the exhaust port and the cylinder inlet coincide with the top of the piston when the latter is at the bottom of its stroke.

The angle *LOI* is that at which the cylinder inlet opens and *LOI'* is that of closing. At *f* is the position of the piston for both opening and closing, and *i* is the width of the port. The distance *d* through which the piston travels between the opening of the exhaust and the opening of the inlet is known as the "drop."

It should be noted that the points of *cut off* are the important ones in the ports, and these are the tops of the exhaust and the cylinder inlet ports and the bottom of the inlet port to the base. So long as the width of the ports is sufficient, the position of the opposite edges has no effect upon the timing. Therefore the important edges of the ports should be finished, which can be conveniently accomplished by milling from the interior of the cylinder. For the same reason the position of the piston-pin hole in the piston should be measured from the top.

The check valve of a two-port engine opens just so soon as the vacuum in the base is sufficient to overcome the resistance of the spring. It will close so soon as the pull of the spring exceeds the vacuum. The time during which the valve will remain open depends upon the speed of the engine and the area of the valve. For the purpose of computation it may be assumed as 120° of crank travel. For speeds up to about 600 r.p.m. the ordinary stock valve, although comparatively heavy, gives good results, but for high speeds the valve should be of large diameter and small lift and very light. Valves of sheet steel working against leather-faced valve seats have been used with good success on small high-speed two-port engines. It is best to make such valves with a lift about one-eighth their diameter, and therefore the diameter of the valve opening should be larger than the required area of the inlet passages in consequence.

In the base-compression two-cycle the base should be designed with a minimum clearance, the travel of the lower end of the connecting rod being carefully laid out for this purpose. The clearance of the inside of the base about the end of the rod and the ends of the crank cheeks should be ¼", but at the sides of the crank cheeks the clearance may be cut down to ⅛".

In the design of the inlet manifold, the distance the gas must travel from the carbureter to any one of the inlet ports should be approximately the same. The travel of the charge should always be upward to the inlet port, so that any condensed gasoline will flow back into the carbureter and not have a tendency to flood the base.

In the design of the exhaust manifold, that portion of the exhaust passage forming a part of the cylinder should be water-jacketed. If this is not done the cylinder will expand unevenly when it gets hot and the pistons will stick. The passages in the manifold should be so arranged that the gases from one cylinder in which the exhaust port is just opening will not discharge directly into the port of another cylinder which is about to close.

A two-port engine which falls off in torque as it approaches high speed may be improved considerably by the use of a small third port in the cylinder wall. This may be connected to a manifold and a carbureter separate from that supplying the base through the check valve, or it may open directly to the atmosphere. With the latter arrangement, the third port should be covered when starting and at low speed. For high speeds the port is opened and at the same time the gasoline needle valve is given a slightly larger opening.

This is the combined type of engine sometimes called the "four-port" referred to in Chap. I. The area of the auxiliary third port should be sufficient to make up the deficiency of the opening through the check valve for the speed at which it is desired to run the motor.

Besides the third port and the check valve there are quite a number of arrangements employed for controlling the admission of the mixture to the base. Of these the rotating tubular valve and the disk valve carried on the crankshaft are the most common. These valves have the advantage that they may be opened at a different angle than that at which they close. For example, if the total duration of the opening of such a valve is 90°, it may be opened at 60° in advance of the center and closed at 30°.

Differential piston motors and those using other methods of supplying the mixture to the cylinder, require special treatment for each individual case, and therefore no set rules can be given for their design. In the Koerting type of two-cycle, in which the air and the gas are each supplied by separate pumps, at a pressure of 8 or 9 lbs., the exhaust port opens at an angle of about 40° in advance of the end of the stroke, and shortly after the piston is 20° from the end of the stroke air is admitted from the air-pump. The fresh air assists in clearing the cylinder of dead gas, and after the piston has passed the center about 10°, the gas is allowed to enter from the gas pump. The air-pump and the gas-pump discharge into the working cylinder after the exhaust port is closed and until the piston has made about ⅓ of its compression stroke. This is shown by the diagram of the Koerting engine, Fig. 73 *B*.

One of the annoying features of the two-cycle is the tendency of the burning gas as it is rushing out of the cylinder to ignite the fresh gas issuing from the bypass to the base, igniting the charge in the base. This causes what is known as a "back-fire" or base explosion. This occurs usually at low speed with a lean mixture or when the spark is retarded. The reason this occurs when the engine is throttled is that the mixture is diluted by the burned gas from a previous charge and burns slowly so that the flame is still present in the mixture when the cylinder inlet port is uncovered. The same effect is secured when the spark is retarded because the ignition has taken place late, and the flame has not had time to reach all of the charge as soon as when the spark is advanced.

This feature of the operation of the two-cycle is annoying and will frequently stop the engine. It can be avoided by the use of any device in the bypass that will extinguish the flame in its attempted passage to the base. The principle involved is that employed in the well-known *Davy safety lamp* for miners, in which a fine mesh screen *surrounds the flame*, effectually preventing the lamp from *-ting fire to* any mine gas with which it may come in

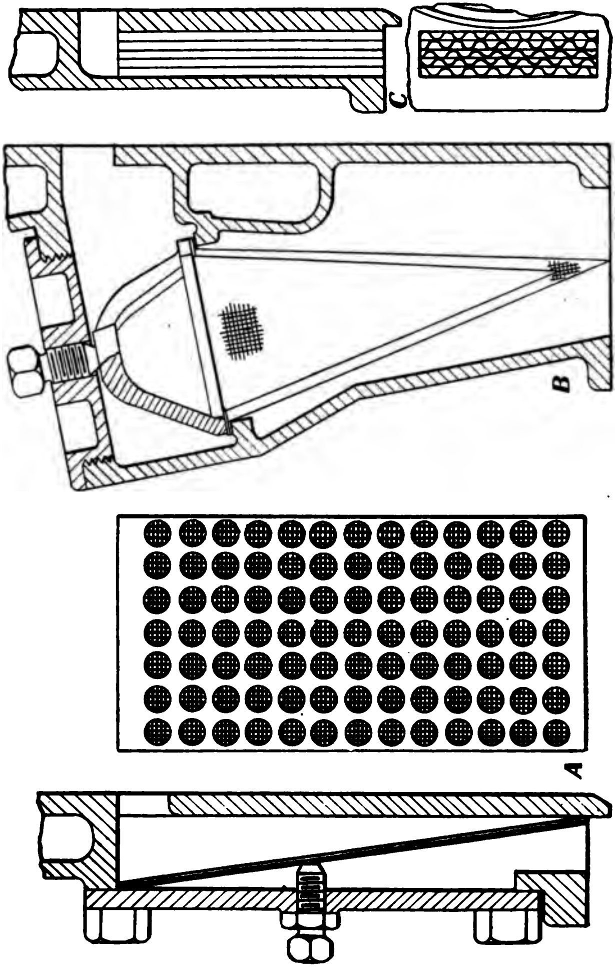


FIG. 74.

contact. The fine screen of the lamp prevents the communication of the flame to the surrounding gas because it chills the products of combustion that pass through it to a temperature below the ignition temperature of the gas.

In the by-pass of many two-cycles a screen is employed, held between perforated plates to prevent it from bursting as shown at *A* Fig. 74. It is placed in a slanting position in the transfer port as shown so that the combined area of the openings will be large enough to prevent choking. Such a screen must be of fine mesh to be effectual, and does not stop back-firing altogether. Furthermore it has a tendency to choke.

Fig. 74 *B* shows a form of screen that appears to give better satisfaction than the flat screen, and to have less tendency to clog. It is placed in a slanting by-pass, as shown, so that it may be easily removed for cleaning.

The form of back-fire preventer shown in Fig. 74 *C* was devised by the author. It is known as the "cellular bypass" and is formed of corrugated plates of sheet metal alternated with flat plates as shown. It has large openings, and the chilling effect is obtained by the length of the tubular spaces. It has been found that the longer the passage the larger may be the openings. This device, when made of proper length and size of mesh, and carefully installed, prevents base explosions altogether. It has very little tendency to clog, and has been in use on some motors for as much as three years without showing any collection of gum. It has the further advantage that heat conducted to the corrugated metal from the cylinder thoroughly vaporizes the mixture passing through it and results in lowering the fuel consumption. The temperature of the plates remains at about 150° F. This temperature is below that of the ignition of the mixture, but above the temperature at which gasoline vaporizes.

CHAPTER XXXI.

AIRPLANE ENGINES.

In the design of a gasoline engine for driving an airplane or dirigible balloon, the engineer should keep in mind three things. These are: To secure, first, the greatest power for the least weight; second, continuous operation for long periods at full speed; and, third, the greatest possible strength within the limits of weight.

It is quite practicable to fly an airplane successfully with a power plant weighing, complete with all accessories, including water and fuel sufficient for one hour's flight, ten pounds or more per actual horsepower delivered. In this total weight, that of the motor proper complete with carbureter and magneto may reach as high as six pounds per actual horsepower delivered. It is desirable, however, to keep the total weight of the power plant as outlined above to within six pounds per actual horsepower, and the weight of the motor proper to within four pounds per horsepower. In fact, much lighter motors than this are made. In the beginning of aviation, catalog weights and horsepowers were not always to be relied upon; the catalog horsepowers running sometimes as high as forty per cent in excess of the actual brake horsepower delivered by the motor at its normal flying speed, and the weights occasionally twenty per cent less in the catalog than the actual weight of the motor.

Continuous operation at high speed, especially when it is gained without falling off in power, requires careful attention to the following: First, the motor should be as simple as it is practical to make it, without loss of efficiency in other ways. Second, if water-cooled, it should have ample water circulation. Third, it should be most thoroughy lubricated. Fourth, every individual piece of the motor should have an ample factor of safety, so there will not be the slightest possibility of a break-down.

The strength of the airplane motor may be made in all its parts even greater than that of either the automobile or the marine motor, by the judicious selection of material having great strength in proportion to weight, and the design in the various sections of forms that will give the greatest strength for the volume of the material employed.

As an example of what has been accomplished in saving of weight in design, the reader is referred to Chapter XXI, last paragraph, on the design of hollow shafts.

The writer wishes to impress upon the designer of an airplane motor the extreme importance of the two factors of strength and reliability. If he will put himself in the aviator's place and imagine the predicament he would be in if the motor were to stop when flying over rough country or over water with an ordinary land machine, he would be more careful in his computations. A forced landing on rough ground or among trees, houses or anything of that character is certain to wreck the machine and result in serious injury to the aviator or perhaps his death.

As examples of what sort of accidents occur to an airplane motor because of faulty design or construction, a few of the early ones will be given. One of the most frequent accidents was the breaking of the crank-shaft. On one motor in particular broken connecting rods were frequent. In one instance the broken rod went through the crank-case and broke the foundation timber. Fortunately, the machine was not otherwise wrecked, else there would have been a fatal result, for the accident occurred at an altitude of 600 feet. Still another motor overheated after a flight of ten or fifteen minutes' duration simply because the designer did not provide either a sufficient water-jacket or sufficient manifolds. Leaky water-jackets on motors with attached jackets were frequent, but not necessarily dangerous. A frequent cause of trouble with some makes of engines was burning of the valves, due to the fact that they were not water-jacketed.

In the selection of materials for the airplane motor, we have fortunately a number of excellent alloy steels which when properly heat treated give tensile strengths running considerably over 100,000 lbs. per sq. in. with high elastic limit and resistance to fatigue. Among these are the nickel steels and those with a small percentage of chromium and vanadium. These are available for connecting rods and crank-shafts and have been used in cylinders of revolving air-cooled motors. Some of these steels will go as high at 150,000 lb. per sq. in. and still have sufficient ductility to be satisfactory. In order to obtain such a result, the proper heat treatment of the steel is imperative. The rule for this treatment should invariably be obtained from the maker of the steel.

There are certain aluminum alloys having low specific gravity combined with tensile strength such as the alloys of aluminum and magnesium which reach as much as 30,000 lb. per sq. in. if cast in a chill. Magnalium, an alloy of aluminum with magnesium and antimony, will show this result on a specific gravity of 2.4. This alloy is available for crank-cases, pistons and other small parts which must be cast. It has been used with some measure of success for cylinders, and, as such, shows good wearing qualities when employed with cast iron pistons. It has the objection for this purpose that it weakens under rise of temperature and may give way if not sufficiently cooled. Crank-cases of this material may be cast as thin as ⅛″ thickness successfully.

For the cylinders of water-cooled motors, cast iron low in combined carbon and comparatively high in silicon may be used with a safe load of 4,500 lb. per sq. in. Cylinders of cast iron should be cast with as small an amount of finish as practicable and this will give a very dense casting and as hard as can be machined. Water-jackets may be made of copper, brazed on or held by shrink rings or may be made of aluminum castings shrunk into place.

The majority of airplane motors are of the four-cycle type. There are, however, very successful two-cycle motors in use both in America and in Europe for airplanes. The latter type has the advantage of simplicity, high power for weight and ease of lubrication. In motors of this type designed by the author, the ratio of weight to horsepower is low and effective lubrication is obtained by mixing oil with the gasoline in the proportion of one quart to five gallons. The oil forces itself into the bearings, which are also provided with grease cups of the automatic type, holding sufficient lubricant for twelve hours' continuous running. These motors have been operated successfully as high as 1,400 r. p. m.

PART III.

OPERATION, TESTING, SELECTION.

CHAPTER XXXII.

INSTALLATION.

A gas engine should be installed in a light dry place with ample room for the operator to get around it on all sides. A cement floor is best, and when laid in the engine room it should be well drained. The top of the foundation may be given a slight inclination endwise for drainage, but it should be level sideways. Always keep the foundation away from the walls and pillars. Never make it smaller than called for by the plans furnished by the maker of the engine, and if the soil is loose make it slightly longer than the plans call for. Be sure to have the foundation laid by a competent mason who understands cement work.

A great many engine builders prefer cushion timbers to cap-stones for stationary engines. Others recommend leveling the engine up with iron wedges and filling in beneath it with a cement mortar of one part cement to one part sand. The easiest method is to build a coffer dam of clay about the engine bed, make the cement very thin and pour it in. After the cement is set, the wedges are removed and the space filled with the cement mortar.

Before bolting the engine to the foundation, it is best to have the connecting rod disconnected from the shaft so that the latter may be easily turned to test for binding. Pull all the nuts down evenly, going from one to another, and giving each a half turn at a time so that they will all pull down together. If one nut is fully tightened while the others are loose, the frame will be pulled out of shape. Test for distortion of the frame by *turning* the shaft in the bearings. If it turns harder *than it did before* tightening the bolts, it is a sure sign that the nuts on the foundation bolts have not been pulled down evenly. The remedy is obviously to loosen one nut after the other until the shaft is freed and then tighten again. If this precaution is not taken the shaft will be sure to heat. Don't deceive yourself on this point by saying that it is good enough and that you will let it go, but keep at it until, after the nuts are pulled up tight, the shaft is free.

All gas engines are designed to "run over." By this is meant that the top of the flywheel turns away from the cylinder, and during the explosion stroke the crank pin is passing through the top half of the crank circle. The belt should, whenever possible, be led away from the engine. To lead it backward or upward puts a strain upon the bearings that they are seldom designed for. The distance between the center of the engine shaft and the center of the driven shaft should be at least fifteen feet on small engines of twenty horsepower and under and longer on larger engines.

Engines of fifteen horsepower and over can be more conveniently started when fitted with a clutch pulley either on the engine shaft or on the line shaft. Clutches are quite a convenience on sizes below fifteen horsepower.

In piping up the engine, make all lines of pipe for whatever purpose as short as possible and as free from turns as circumstances will permit. Use paint, graphite or white lead on the joints of water or gas pipe, but never on gasoline pipe. On the latter use common brown soap or shellac varnish. It is a very good plan to use unions frequently, as it makes it more convenient to take the piping down for alterations or repairs.

It is especially important that the exhaust pipe be free as possible from elbows as they have a decided choking effect and will cause back pressure and loss of power. It is a good plan to avoid common pipe elbows when making a turn and to use pipe bends or two 45° elbows instead. Install the muffler or exhaust pot as close to the outlet as it can be placed conveniently, but place it, if possible, outside the building. Allow a portion of the outlet water to enter the exhaust pipe at a point about one or two feet from the exhaust valve casing, but so that there is not any chance of the water flowing back to the valve. If the exhaust passes upward from the engine, the water should not enter the exhaust pipe. Provide a drain for the water at the lowest point of the line and be careful to avoid traps for the water to settle and choke the passage. In the case of a long exhaust pipe, there should be a drain at the lowest point in the pipe to carry off the water of condensation, whether water is allowed to flow into the pipe or not. Never use smaller pipe than called for in the plan and avoid long stretches of horizontal pipe. Do not allow the muffler to hang from the pipe, but furnish it with a separate support, unless it is very light. Provide for expansion by means of a slip joint, especially where the pipe is long. Under no circumstances lead the exhaust into a chimney or into a drain. When the exhaust pit is used it should be designed to withstand the pressure of a possible explosion. See Chapter XXIX.

Do not install a gas line without some form of regulator such as a rubber gas bag. On high pressure lines such as are found in natural gas regions there should be a pressure reducing valve to bring the pressure to three or four ounces. Be sure that the gas bag or other form of regulator is installed as close as possible to the inlet casing. The purpose of the gas bag is to bring a large volume of the fuel close to the inlet, since, if the gas should be drawn through a long pipe, the pressure would be reduced. The volume of the bag should be at least one-fifth the piston displacement of the engine.

Gasoline piping should be ⅛″ pipe size for engines up to twenty horsepower and ¼″ for large sizes when the gasoline is piped from under ground. For gravity feed when the length of the pipe is ten feet or less, ⅛″ pipe will usually answer for all sizes within the limit of power for which gasoline is used on stationary engines. Insurance regulations require that not over five gallons of gasoline may be kept within a building and therefore when the tank is larger than five gallons capacity it should be placed under ground not less than five feet from the wall of the building. With the tank below the engine, gasoline must be pumped. It is pumped to a constant level chamber on the engine and flows over into a return pipe after reaching a certain level. The tank should be placed not over four feet below the level of the pump.

Under-ground tanks should be of galvanized steel painted with pitch or asphaltum on the outside. The filling tube extends to the top of the ground and should be fitted with a lock cap. Placing the tank below ground has the advantage that it keeps the fuel cool and avoids waste by evaporation.

The air pipe on the smaller sizes of motors is generally led from a compartment in the base. If the engine is installed in a dusty place, it is best to draw the air from outdoors or from a box covered with screens or some form of air filter. Be sure to avoid sharp turns in the suction pipe and, if the pipe must be long, it should be one size larger than the opening into the valve box. Air mufflers for the intake are described in Chapter XXIX.

The cooling of stationary engines is by means of water, except in the very smallest sizes. The water supply may be taken from the water mains and allowed to flow through the water-jacket, or it may flow by thermo-siphon circulation through the jacket and a large cooling tank, or some form of radiator may be employed through which the water is either pumped or allowed to flow on the thermo-siphon system.

When the water is supplied to the engine from the mains, the supply required depends upon the temperature of the water supply. For average conditions the flow should be about 5¾ gallons per horsepower per hour. This is based on the average working load on the engine. As water will generally begin to deposit a sediment in the water jacket at 185° F., the temperature of the outlet water should be kept just below that or at most 180° F.

At full load, the heat carried away in cooling is about 1⅛ times that absorbed in power delivered, and this will give the engineer a basis of computing from the known temperature of the water supply and that of the outlet water, the desired rate of flow. The higher the temperature range (difference between inlet and outlet temperatures) the smaller may be the flow.

For plain cylindrical tanks, cooling the engine by thermo-siphon circulation as shown in Fig. 75, the size of the tank is given by various authorities at from 20 to 100 gallons per B. H. P. The average practice is to supply 50 gallons per B. H. P. The water should be piped from a point about four inches from the bottom of the tank to the lowest point on the water-jacket of the engine and

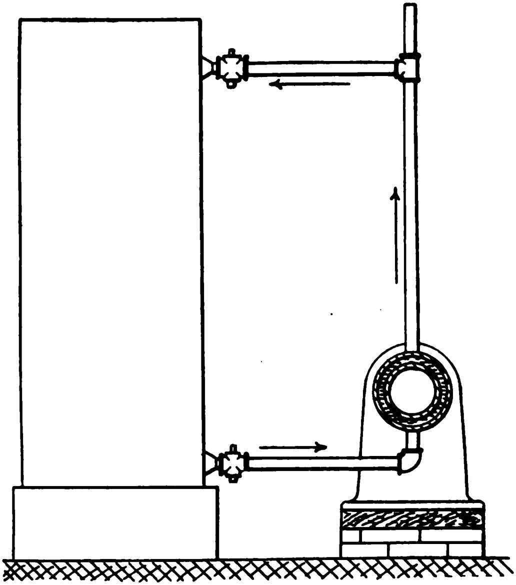


FIG. 75.

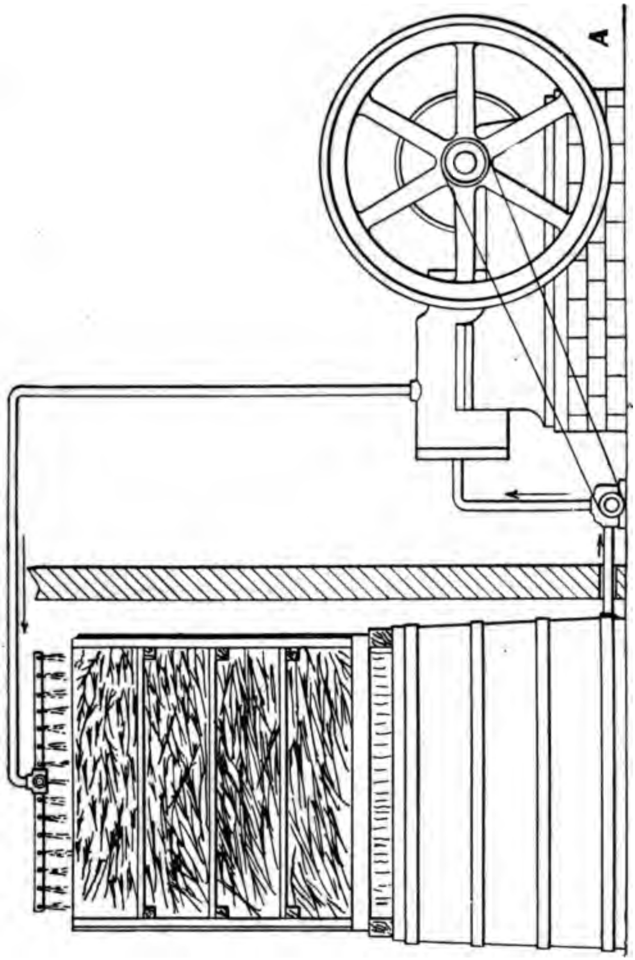


FIG. 76A.

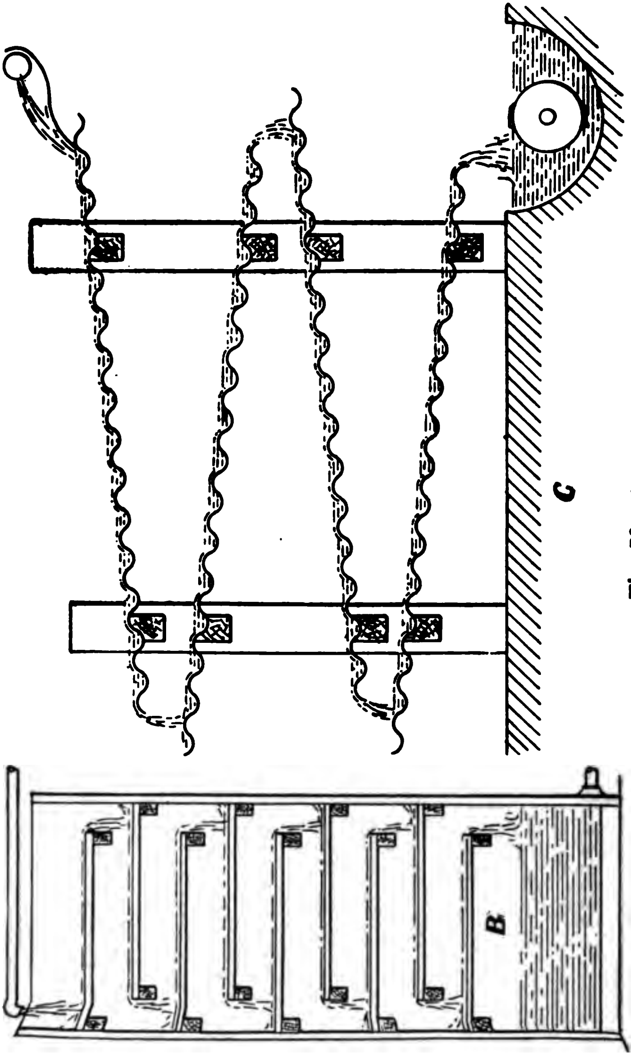


Fig. 76. A and B.

from the highest point of the water-jacket to a point a foot to two feet from the top of the tank according to its size. The water should at all times be above the outlet, usually from 10″ to 14″ higher. The amount depends upon the rate of evaporation of the water. If there is a water system where the engine is installed, the water should be piped to the top of the tank and a valve placed in a convenient position for filling. An ordinary water-closet float valve will be found convenient, as it may be arranged to keep the tank filled automatically.

For cooling the jacket water of engines where the room taken by a large tank is not available, or where the engine is so large that a cooling tank of sufficient capacity would be cumbersome, various cooling appliances are available. One is the radiator of the same general style as that employed on an automobile. Radiators are made in various forms, some with plain tubes, others with fin tubes and still others are employed for stationary and portable engines much on the same plan as the upright tubular boiler. The latter is usually fitted with a fan, either below or above the tubes. In fact, the common steam radiator has been employed. Various forms of cooling towers are in use. In one the water is allowed to flow from the outlet over layers of tin scrap or brush through which air is allowed to circulate freely, as shown in Fig. 76 *A*. Another form of tower is made of broad shelves over which the water flows back and forth as shown in Fig. 76 *B*. A third form of cooler is shown in Fig. 76 *C*. It is made from corrugated iron arranged as shown. It should be noted that for a cooling tower, a circulating pump is required. The cooling is effected in devices of this kind, partially by contact with the air and partly by evaporation.

Still another method of jacket water cooling may be found a convenience in certain localities. This is the use of a large, but not necessarily deep pond into which the water is allowed to flow to the top and the cool water pumped from the bottom, preferably from a point at some distance from the hot water discharge.

The following data will be found useful in getting the proper sizes of various pipes and appliances. The *area of the* inlet water pipe for forced circulation should *be at least* .022 square inches per H. P. Smaller pipe than ½-inch is seldom used. For the outlet it is customary to use one size larger pipe than for the intake.

For gas pipe where the average city gas is used, the area of the pipe should be at least .1 sq. in. per H. P. The meter should have a capacity sufficient to take care of the gas required for the engine at its maximum power. Meters are usually designated by the number of lights they will serve. The capacity of a meter is usually about five cubic feet per hour per light. Thus a five light meter will handle about 25 cu. ft. of gas per hour, a 100 light meter 500 cu. ft., etc.

CHAPTER XXXIII.

STARTING AND STOPPING.

Starting a gas engine is not a complicated or a difficult operation, if it is gone about properly. Not all gas engines start on the same or similar adjustments such as the position of the gas valve, setting of the igniter, or amount of gasoline used to prime them. The principle involved is, however, the same in each.

When a gas engine has been standing idle for some little time since it was last in operation, it may be that some of the parts are stiff. Make certain that all valves and parts pertaining to them are free. Fill all oil cups, even if they are partially full already. If grease cups are used on any of the bearings, see that these, too, are well supplied with lubricant. Turn on all oil cups. If the engine requires priming, fill the priming cup with gasoline, and with the flywheel turned until the piston is in the position to begin the suction stroke, open the cup and allow the gasoline to flow into the cylinder. Then, with the priming cup still open, turn the flywheel until the suction stroke is nearly completed, and close the cup.

Two methods of starting are now available, depending upon the size of the engine. If the engine is small enough, the igniter may be set at retard and with the switch closed and the gas valve in the starting position, the flywheel may be turned quickly in the direction the engine should run. One, or at the most, two revolutions past the ignition point should be sufficient to start. So soon as the engine has started, gradually turn the gas valve to the running position and advance the ignition.

For large engines which cannot be turned past the compression by one man, proceed as follows: After priming and drawing in a charge of air, close the priming cup, and pull the flywheel quickly backwards as far as it will go, and just as the movement is checked by the compression, snap the igniter or in some other way ignite the charge. The resulting explosion should be sufficient to drive the engine through four revolutions or two cycles. After the engine starts, open the gas valve promptly to the running point and advance the spark.

Should this method fail to start the engine the first time, close the gas valve and, with the priming cup open, turn the engine seven or eight revolutions to clear the cylinder of excess gas before priming again. It may require a little practice to judge just what amount of gasoline to use for priming. Usually the priming cup will hold just enough for one prime, but this is not always the case.

In some cases the engine is supplied with a relief cock that will allow only a portion of the charge to escape, leaving enough for starting. Other engines are provided with a double cam on the exhaust for pressure relief.

After the engine is started, turn on the water at once and see that all the oil cups are feeding the right amount. After the engine has picked up its load, adjust the water supply to the proper temperature.

For starting on compressed air, two general plans are employed. The average engine of the single cylinder pattern is fitted with nothing more than a hand operated valve. With a valve of this kind, the flywheel is turned until the piston is in the position for beginning the expansion stroke and slightly past the center. The switch is closed and the gas valve is set to the starting position, while the igniter is set to retard. With a quick movement the air valve is opened wide, starting the engine, and it is closed quickly after the piston has completed from one-half to about two-thirds of its stroke. The compressed air is exhausted during the next stroke and the engine should then pick up its cycle.

On other engines, and especially those having more than one cylinder, a special air-starting valve-gear is employed. Quite often this is fitted to one cylinder only while the others are arranged to take up the cycle after getting under way with the air on the cylinder having the air gear. In this style of engine, all that is needed is to operate the starting lever provided for the air gear, get everything ready and turn on the air, then the engine will take care of itself. On some engines the air-valve gear is arranged to throw out automatically and no further attention is required. On others the starting gear must be thrown out by hand. Priming with compressed air starting is not usually required.

A little practice with the ordinary gas engine will enable the operator to start on gas without priming with gasoline. By trial, a point will be found at which the gas valve may set so that the engine will draw in just the right amount of gas, when turning the engine by hand.

For starting a gasoline engine provided with a pump for the fuel, it is necessary to operate the gasoline pump by hand a few strokes, in order to fill the small reservoir from which the gasoline is drawn into the intake. Usually the running position of the needle valve is all right for starting. Some engines require a little larger opening for starting, and the needle valve is then gradually closed as the engine warms up. Priming usually makes the starting of a gasoline engine easier than starting with the gasoline picked up from the vaporizer. With engines employing a float feed type of carbureter, the float is depressed when starting by means of the flushing pin, and a small amount of gasoline is allowed to flow into the air pipe.

Always, before attempting to start an engine, see that everything is in good order. Examine the igniter and all parts of the engine that may give trouble, such as loose wires, lack of gas or gasoline, etc. See that the water circulation is all right, and where tanks or similar coolers are used examine them to see if they have sufficient supply of water.

In stopping, open the switch and turn off the fuel, either gas or gasoline, then turn off the cooling water and stop the oilers feeding. In freezing weather, always drain all the water out of the jackets as well as from other parts of the circulating system.

CHAPTER XXXIV.

CARE OF GAS ENGINES.

The proper care of a gas engine should be the pride of any engineer who has one in charge. In every shop or factory where a gas engine is in use, it should be left in the care of one certain man, and that man should be held responsible for the good condition of the engine at all times. Do not leave the engine in the care of any man who happens to be about when it needs attention. If no one is responsible, no one will take an interest in the engine, and it will be shut down frequently for want of some attention that has been neglected.

Above all things keep the engine clean, well oiled and with an ample supply of water at all times when a tank or other closed circuit cooling system is used, or with a sufficient flow of water, if connected to a pressure system. Careful and frequent oiling is necessary on the joints of the governor to keep it working freely and from racing, and also on the joints and bearings of hammer-break igniters.

If the engine is new when set up, and the timing of the valves and the igniter are not marked on the flywheel, it is best to mark them. Mark the flywheel with a prick punch or a chisel on the side of the rim at a point opposite the most convenient stationary point on the engine, and with a steel stamp or the point of a punch, letter each mark. The positions to mark are as follows: The point at which the igniter snaps in the running position; the time of opening and the time of closing of each of the valves.

The best way to determine the point at which a valve is closing is to place a piece of paper between the push rod and the end of the valve stem and to turn the engine over slowly until the paper is gripped. Mark this point and then, turning the engine over in the running direction, note the point when the paper is released. This is the *point of valve closing.* All mechanisms wear more or less and these marks will be found a great convenience when there is occasion to set the valves or the igniter. In fact, it is a good plan to run over the settings every few months, to see whether appreciable wear has taken place. Always run over the valve settings after regrinding the valves.

In marking the valve settings or in setting the valves to marks already made, be sure that the point is not passed. If it is, the engine should be backed up about a quarter revolution and then turned ahead again to make sure that all lost motion has been taken up. If there is no convenient point upon the frame work of the engine by which to locate the mark on the flywheel, drop a plumb line across the center of the shaft or across any convenient point and mark by the line.

In spite of the fact that there are more oils advertised for the lubrication of gas engines than for any other purpose, there are some who persist in using oils that are not at all suited to this purpose. Never use the heavy oils that are sold for steam engine cylinders and do not use animal oils for this purpose. Specially refined castor oil is used for air-cooled airplane motors, but it is too expensive for ordinary use. Oils for gas engines are specially refined so as to be free from carbon and are usually thinner than oils for steam engines. Don't judge the quality of an oil by its price. For the average stationary engine a medium priced oil is quite often good enough, but for high speed marine or automobile engines, the selection of the proper oil to suit a particular style of engine usually requires quite a little experimenting. It is usually quite safe to follow the makers' recommendations in the matter of the proper oil. For the bearings, a good grade of machine oil will suffice on a stationary engine, but generally, the same oil as for the cylinder is employed and the buying of two grades of oil is avoided in this way.

It is a good plan to examine the various parts of the engine at regular intervals. The exhaust passages should be cleaned every four to six months, especially if the fuel employed shows a tendency to clog the passages with soot. Clogging of the exhaust passages causes back pressure and reduces the power of the engine. The cooling water, especially when heavily charged with lime or carrying sediment is likely to clog the water-jacket. This is cleaned most readily by means of a long iron rod bent at one end and similar to a stove poker.

Examine the valves occasionally, paying particular attention to the exhaust valve, which is the one that leaks most frequently. On the first sign of leaking, a valve should be reground at once. The springs should have attention as they occasionally weaken and fail to pull the valve to its seat properly. Watch them to see that they follow the push rod or other operating device promptly. See that the exhaust valve is not pulled from its seat by the suction when the engine is running throttled. In case this occurs, provide a stronger spring. A suction inlet valve should respond promptly to the suction and yet close firmly. The stems of the valves should be oiled, the stem of the exhaust valve with kerosene and the others with cylinder oil.

Keep careful watch of the igniter, and clean points at intervals of one to four weeks according to the nature of the fuel. Watch the parts for wear and the springs for loss of tension. When a battery is employed, have a spare on hand always, and have the spare battery connected to the line by a two-throw switch so that it may be put in service instantly. When a magneto is used for ignition, it is a good plan to have a battery on hand for emergencies.

Should it happen that the engineer has forgotten to turn on the jacket water, and the engine shows evidence of heating by throwing smoke through the open end of the cylinder, the water should be turned on cautiously. If the water is turned on full volume at once, the cylinder will contract before the piston has had a chance to cool and the piston may seize. At the same time the cylinder oil cup should be opened full for a few moments.

Feel of the bearings occasionally and keep close watch on the oil cups, watching the sight feed and also the rate at which the supply of oil falls in the cups, thus keeping tab on the rate of feed in case the sight feed gets clogged. In case of a hot bearing, do not attempt to cool it too quickly, as with an application of ice, or the *bearing may seize* the shaft. In case ice is at hand, apply *it to the shaft itself* and the shaft will contract away *from the surrounding* bearing.

Keep the governor working freely and carefully adjusted. If the governing is by hit-or-miss, see that the gas valve opens promptly on the governor getting into operation, and that an explosion is not missed waiting on the sluggish action of the governor.

Take a look at the nuts on the cylinder head every few days when the engine is new and tighten them up, if they are loose. The bearings will need occasional adjustment when new. After a time the shafts will set well in their bearings and require no adjustment for long periods. In taking up bearings, be sure to have the edges of the boxes held firmly against each other. The adjustments should be made by means of shims, small pieces of sheet metal or paper placed between the two halves of the bearing to keep them the correct distance apart. Be sure to remove the same thickness from each side of the bearing so that it will be drawn up evenly. After removing the shims, draw the bolts down tight. If this causes the bearing to pinch the shaft, replace just enough to allow the bolts to be drawn down snugly without binding the shaft. This applies to both the shaft bearings and the connecting rod bearing on the crank pin.

Never run the engine with any of the bearing bolts loose and be especially careful on this point with the connecting rod. To test the fit of the crankpin bearing, pull the flywheel back and forth through a small arc and note if the rod follows the movement promptly. If the rod does not move with the crank, it is too loose.

The piston pin bearing seldom requires adjustment. For this reason, small engines quite frequently have this end of the rod fitted with a solid bushing and on many stationary engines the brasses are fitted with the edges touching and without shims. In this case of the solid bushing, the only adjustment is to renew the bushing. When the halves have no shims, it is necessary to file the faces of the brasses.

Be sure to replace all locking devices, either jamb nuts, lock washers or cotter pins as the case may be, before starting the engine. Always attend to a loose bearing at once as the tendency is for a bearing to get worse in this respect if neglected.

In the care of the igniter, the most important thing is to keep the points clean where they come together. In case they become corroded, they set up too much resistance in the path of the current and a poor spark results. The points may be trimmed with a fine file or fine sandpaper. A small flat Swiss file passed between the points while held together is most convenient. Another way is to use a narrow strip of sand paper (No. 0) doubled, with the sand out, and to pass this back and forth between the points. The metal should be bright and surfaces touch all over.

In average service, the exhaust valve will require attention about three times a year. If the valve is seating well, the surface of the seat will show bright and in contact both on the valve and on the seat. Blackening of either, indicates that grinding is necessary. This should be done with fine emery mixed with oil. In the head of nearly all valves there is a slot which may be used to turn the valve on its seat. The most convenient tool for this purpose is a screw-driver bit held in a brace. The valve should be turned first in one direction and then in another, back and forth, to secure the best results. Finish with flour emery, and clean the emery out thoroughly with kerosene or gasoline. If there is any chance of the emery getting into the cylinder, the opening to the cylinder should be stopped with waste before grinding is begun.

At intervals of six months, if the engine is in constant use, remove the piston and clean it and the inside of the cylinder with kerosene. Remove the rings from the piston and give both the rings and the ring grooves in the piston a thorough cleaning. Scrape off any deposit of carbon both from the piston and the cylinder head.

The accumulation of carbon may be due to poor cylinder oil or an excess of either oil or fuel. With large engines the deposit is usually scraped out. With small engines, burning out the carbon with the aid of oxygen is frequently resorted to. Water admitted with the air or the mixture will not only frequently remove a carbon deposit, but will keep it from forming. When using fuel oil with an appreciable sulphur content, water should be avoided, as it tends to produce sulphurous acid, which will attack the cylinder walls.

CHAPTER XXXV.

TROUBLES.

The enumeration of every possible source of trouble that might occur in the operation of a gas engine is out of the question, but a general outline of those of most frequent occurrence will be given. These will, the author believes, give a clue to the solution of problems not considered in this chapter.

Failure to Start.—It is a good rule to examine the ignition apparatus first of all. Test this out systematically, beginning at the igniter or the spark plug, but first of all see that the switch is closed. On a hammer-break, remove the wire from the stationary or insulated electrode and rub it on some part of the engine. If current is flowing to the igniter a spark will be produced. If the spark is not strong, it is most likely that the battery is weak, or some connection in the circuit may be loose, but not disconnected. If no spark is obtained, trace up the connections and, if they are all tight and there is no break in the wiring, examine the battery.

If a spark occurs when the wire is tested, the trouble may be in the igniter itself. First force the points of the igniter together and draw the wire across the end of the insulated electrode. If no spark occurs, either the points are not meeting or they are corroded. If a spark does occur when the points are held together and the spark is obtained also when the points are forced apart, there is a short circuit between them and the current is flowing across from one to the other where it should not. Either the insulation has broken down and requires renewing or the bodies of the electrodes are touching, or some metallic contact has been made that requires adjustment. Perhaps fouling of the igniter has caused a short circuit.

In case jump spark ignition is used, the wire should be removed from the plug and tested for spark by holding it close to but not in contact with some part of the engine. A weak spark or lack of spark indicates trouble on the act as an ignition point, or carbon will collect in the form of a small projection and do the same. Some forms of spark plugs with long electrodes or with the points projecting too far into the cylinder will cause prematures. *Back firing* is the explosion of the gases in the inlet passages firing back through the carbureter. Its action and cause in the two-cycle is fully explained in Chap. XXX. In the four-cycle it is usually due to a weak mixture, and occasionally to the too early opening of the inlet valve. If the inlet passages are so situated that they get too hot, back firing will occur. Again the entering gas may meet an overheated point in the cylinder, and take fire from that.

Spark Gradually Weakens.—Nearly always indicates a battery too far run down to be of further service. In the case of a dry-cell battery, the life of the battery may be lengthened a few hours by boring a hole through the wax and adding water. A magneto that dies down shows poor magnets, and it should be recharged or exchanged for a new one.

Engine Pounds.—Loose connecting rod bearings, prematures or a loose flywheel key will cause an engine to pound.

Engine Does not Develop Full Power.—See *Weak Explosions* and *Engine Slows Down.*

Smoke.—A white smoke issuing from the exhaust indicates excess of oil, a black smoke excess of fuel. Smoke issuing from the open end of the cylinder may be due to a leaky piston or one that is overheated. The leaky piston may be due to an eccentric cylinder or to stuck or worn rings. Too rich a mixture or faulty ignition may cause smoke from the cylinder.

CHAPTER XXXVI.

TESTING.

Every manufacturer of gas or gasoline engines has a department known as a testing floor or testing room, which is equipped more or less fully with apparatus for determining the performance of the engines before they leave the factory. The testing arrangement of the average factory is usually such that the engine may be run for a short time by means of external power, then on the power of the engine running light, and finally on a gradually increasing load ending up with a full power load for several hours. As the engine is usually driven by means of a belt during the first period, this operation is known as "belting in."

Quite a number of arrangements for a factory test room are employed. A very excellent test room for stationary engines is fitted with a number of electric motor-generators, to which the engines are belted for tests. Each generator has its individual switchboard, and the preliminary belting in is accomplished by running the generator as a motor. The power of the generator is then absorbed by the generator supplying current to a bank of lamps or supplying the current to other generators for belting in other new engines. The power is measured by suitable instruments on the switch-board. While such a testing plant is expensive, it is the very best way in which to test not only the power of the engine but the efficacy of its governing mechanism as well.

A certain testing plant for a marine engine factory is arranged as follows: A tunnel or trench about four feet deep is fitted with a series of shafts at right angles to the center line of the tunnel. These shafts extend alternately each side of the main tunnel in shallow branch tunnels which pass just beneath a block of concrete carrying an iron test block for the motor. On each end of, these shafts, is a pulley, one in the main tunnel and the other at the outer end of the branch tunnel. The inner pulleys are in line and a driving belt passes over and under them, as shown in Fig. 77 at *A*. In the same figure at *B* is shown the manner of belting the engine by means of a belt passing around the flywheel of the engine and the pulley *p* on the end of the driving shaft. The belt is tightened by means of a very heavy pulley *t* held against the belt by its own weight. Over the main tunnel is a large tank of water in which are propellers, one for each testing stand. The propellers are so chosen that the engine must drive the propeller at a certain speed to give the full power rating. Another convenience in a testing room of this character for small engines, is a flat rail trolley passing over the testing stands and thence into the assembling room.

In the average testing room of a modern factory it is customary to carry on only such tests as are required to determine if the engines are up to their standard in power and in proper adjustment. For research work such as the testing out of new designs and new devices submitted by accessory concerns, there is usually an experimental room fitted up with instruments such as dynamometers, for determining the power given off by the engine, indicators or manographs for indicating the engine, speed indicators, meters of various kinds, etc.

The determinations to be made in any test of a gas engine depend to a considerable extent upon the purpose for which the engine is to be used. For a stationary engine the main points to be determined are the fuel economy and the efficiency of the regulation. For the automobile engine, we have power for weight, flexibility of control and fuel economy. For heavy duty marine engines fuel economy is first and flexibility of control a close second. Power for weight is of minor importance. In medium and high speed marine engines for small boats, power for weight and flexibility are as important as fuel economy. For racing boats power for weight is of first importance, fuel economy second and flexibility third. For airplanes power for weight is the prime factor, fuel economy and flexibility are desirable but far less so than light weight and reliability at high speed.

The power derived from any fuel is determined by that proportion of the heat developed by its combustion that may be transformed into work. The most efficient engine is that in which the greatest amount of the heat of mbustion may be turned into delivered horsepower. That

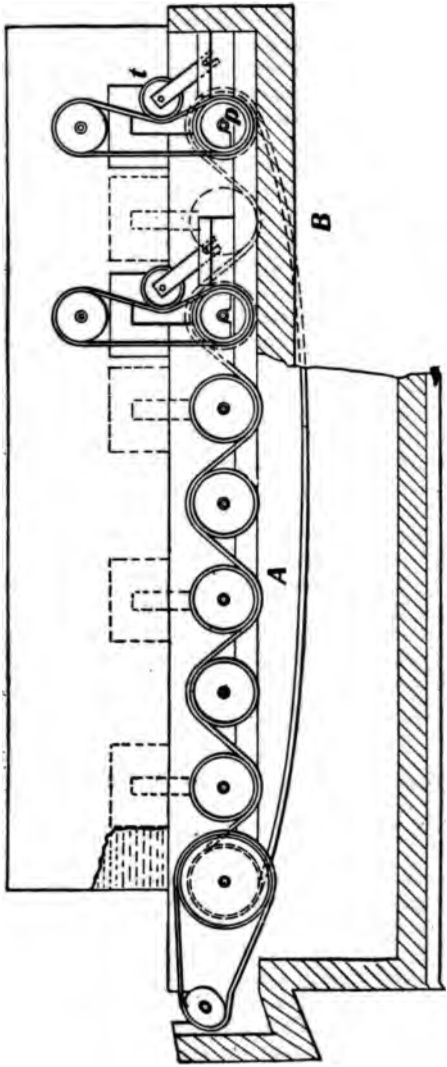


FIG. 77.

portion of the heat not transformed into work is considered as lost, and it is the determination of the nature of these heat losses and the quantity of each that forms the principal part of the *complete* test of a gas engine. Assuming complete combustion of the fuel, the heat not turned into work is lost through four different channels: heat carried off by the cooling water, heat carried off by the exhaust gases, heat lost by the friction of the engine itself, and heat lost by radiation.

The first three of these items may be measured, while the fourth is considered as the unmeasured difference. In many tests, on account of the difficulty in measuring the heat loss by the exhaust gases, this item is coupled with the radiation loss and the heat unaccounted for by work, cooling and friction is considered as that lost by exhaust and radiation.

In order to obtain a complete heat balance, it is necessary to determine the exact amount of heat given off by the combustion of the fuel. This is determined by means of a piece of apparatus known as a calorimeter. In this device a given quantity of the fuel is burned in such a way that all the heat given off is absorbed by water, the quantity and the rise of temperature of the water both being accurately measured.

In Fig. 78 is shown an example of a gas calorimeter so designed that the determinations of the heating power of the gas can be taken during the test of an engine. Running a test of the gas at the same time the engine is being tested, disposes of any question of the accuracy of the determinations. In the figure, which shows the calorimeter in section, the gas flows to the burner *s* through a very accurate meter *M*, and the burned gas passes first upward through the central flue *n* and thence downward through a number of annular spaces *p* surrounded by flowing water. The path of the products of combustion is shown by the arrows with feathers, and the path of the water by the plain arrows. The cooling water enters at *a* and flowing past the thermometer *d*, passes downward through the spaces *e*, thence upward through the spaces *u*, from which it flows around the spaces *v* between the gas channels *p*, and finally the water enters the space *h*, passing the thermometer *j*, and thence through the overflow *k* to the graduate *m*. The final path of the products of ~~~nbustion is through the outlet q and in passing out,

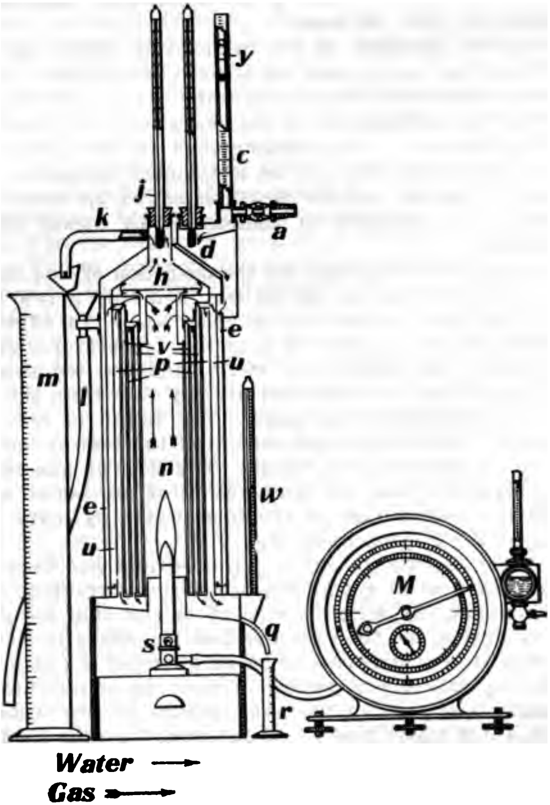


FIG. 78.

the gas flows over the thermometer *w*. Beneath *q* is a small graduate *r* to measure the water formed by the combustion of the hydrogen in the gas. At *c* is a water level tube containing the float *y* to enable the operator to regulate the flow of water.

In the operation of the calorimeter, there are four determinations to be made for a given flow of gas. These are the temperature of the inlet water read on the thermometer *d*, the temperature of the outlet water by means of the thermometer *j*, the measurement of the amount of water which flows through the instrument during the combustion of the gas and the measurement of the water contained in the products of combustion by means of the graduate *r*.

Complete instructions for the operation of this instrument are furnished by the makers, and only a few brief suggestions for its use will be given here. In adjusting the instrument, it is best to have the temperatures of the inlet water, the gas at the meter, and the temperature of the products of combustion as they flow from the outlet *q* approximately the same. The latter is very important in order that entire amount of the heat of combustion be absorbed by the water. The flow of gas should be so regulated that the temperature of the outlet water should not exceed that of the inlet water by more than 10° C. or 20° F. approximately.

After the instrument is adjusted and has been run a sufficient time to get all the temperature readings close to a constant, the water is allowed to flow into the graduate *m* during the time the hand of the meter is passing over say, twelve divisions. Just as the hand is passing O, the tilting funnel *x* is swung to pass the water into the graduate, and just as the hand passes 12, the water is deflected into waste pipe *l*. At the same time the readings of the thermometers *d* and *j* are taken.

Each division of the ordinary experimental meter designates 1-600 cu. ft. so that 12 divisions should represent 1-50 cu. ft. The graduate for laboratory work is usually graduated in cu. cm. and the thermometers in tenths of a centigrade degree. Therefore the fraction of a litre of water multiplied by the difference in the temperature of the outlet and the inlet water gives the number of calories of heat given off by the gas. To reduce this figure to B. T. U. multiply by 3.97. Dividing this result by the fraction of a cubic foot for which the determination was taken, gives the B. T. U. per cu. ft.

This may be more clearly understood by the following example. Suppose that the average of three readings showed for twelve divisions of gas, the inlet water thermometer showed 13° and the outlet 19.8° and that water measured 426 cu. cm. we would have

Calories=(19.8—13) × .426=2.8968 say 2.90

Since this is for 1-50 cu. ft., we have for one cu. ft.

2.90 × 50 = 145 calories, and

145 × 3.97 = 565.65 B. T. U. per cu. ft.

For the purpose of the ordinary gas-engine test, no correction need be made for the temperature of the cooling water, but for accurate work this correction should be made. For an extensive series of determinations time may be saved by having the thermometers graduated in tenths of a degree Fahr. and weighing the water. This eliminates the conversion from calories to B. T. U. and the necessity for correction for the temperature of the water.

This determination gives what is known as the *high value* for the heat given off by the combustion of the gas. In the calorimeter, the steam formed by the combustion of hydrogen is condensed into water. In condensing from steam to water the steam gives up its latent heat and a deduction from the previously determined high value should be made in the proportion of .6 calorie or 2.382 B. T. U. for each cubic centimeter of condensed products collected during the consumption of the unit quantity of gas. This gives what is known as the *low value* for the heat given off by the combustion of the gas.

Some authorities recommend that the determination of the heat balance of gas engine be based upon the high value of the gas. As the temperature of the exhaust gases is always above the boiling point of water and the water formed escapes as steam, the author believes that this rule is not justified and that the low value should be taken as the basis of the heat balance. In any case, the report of the test should indicate which value is taken.

For the determination of the heat value of liquid fuels a special lamp arranged on the pan of a delicate balance is used in the place of the bunsen burner and the meter used for gas. In the use of the lamp, the balance is set just a little under the weight of the lamp and just as the weight is balanced the water is started into the graduate. The balance is again set for a still lighter weight and the water turned into the waste when the scale comes to balance once more.

The test of a gas engine is seldom complete without a determination of the heat value of the gas. Usually fuel consumption guarantees are based on a gas of a certain heat value in B. T. U. per cu. ft., and in case of a controversy the engine must be tested and not only the fuel consumption per horsepower, but the heat value of the fuel itself should be taken coincident with test of the engine. In the absence of the calorimeter, samples of the gas should be taken at frequent intervals during the test and then taken to the nearest laboratory for determination of the heat value. In case the data is to be used as testimony in a law suit, the test of the gas should be made in the presence of the engineer who is to testify, or a sworn affidavit should be made by the laboratory operative who makes the test.

In the case of a consumption guarantee of an engine using liquid fuel, the heat value of the fuel is rarely mentioned, but only the consumption of so much fuel of a certain specific gravity in degrees Baume.

The heat transformed into work upon the engine piston is determined by means of a good steam-engine indicator fitted with a piston having an area of ¼ sq. in. and if possible, a pencil movement of extra strength. The modern type of indicator with an outside spring, is very well adapted to gas-engine testing as, with this type, the spring is not subjected to the high temperature of the combustion of the gas. For the indicator, some good form of reducing motion should be employed in order that the motion of the stroke of the engine may be reduced to a movement of between 2½″ and 3½″, which is the usual length of the indicator diagram.

It is quite difficult to use the ordinary steam-engine type of indicator on engines running at speeds of over 300 r. p. m., and for high speed engines there has been devised a special type of instrument, known as the manograph. This instrument does not employ a piston and a spring for the determination of the pressure within the cylinder, but instead it makes use of a steel diaphragm against which the pressure within the cylinder impinges. The movement of the diaphragm is very slight and, in order to amplify it, the motion is transferred to a mirror, which is, at the same time, given a rocking motion in a direction at right angles to the movement caused by the pressure. The mirror is arranged to reflect a powerful beam of light, either on a ground glass screen or on the film of a photographic plate. By careful adjustment, a diagram similar to that obtained by the steam-engine indicator is secured.

There are several forms of this instrument, of which one, the Hospitalier, is shown in Fig. 79. Three views of the instrument are shown. At *A* is a section taken through the device on a horizontal plane, with the source of the light at *a*, deflected by means of the prisms *c* on a mirror *d* and thence to the ground glass plate *e*. At *B* is shown the arrangement of the mirror support, and *C* is a section taken horizontally through the diaphragm. The mirror is held by the spring *s* against a triangular plate *t*. The point *q* of this plate remains in a fixed position, while against the point *r* is one end of the rod *f* and against the point *o* is one end of the rod *j*. The opposite end of *f* rests against the steel diaphragm *g*, which rocks the mirror in a vertical plane in accordance with the pressure in the tube *u* communicating with the engine cylinder. The opposite end of the rod *j* rests against the end of the lever *k*. This lever is given a reciprocating movement by means of the small connecting rod *l* connected to a pin on the gear *m*. This gear is driven through the gear *n* having an equal number of teeth to gear *m*, and driven by means of a flexible shaft from the engine. Gear *n* is supported on a disk *p*, which may be rotated by means of a worm wheel *w* and a worm *x* controlled by the knob *v* to bring the movement of the rod *j* in phase with that of the engine piston.

This device will deflect the beam of light horizontally according to the movement of the engine piston and vertically according to the pressure in the cylinder. In this way a diagram is shown on the screen similar to that obtained from the indicator. By substituting a photographic plate for the screen, a permanent record of the diagram may be made.

The author will not attempt to give instructions for the use of this instrument, for the reason that carefully written instructions accompany it, and quite a little practice is required to secure accurate results. It has proven itself of great value in the study of high speed engines.

The heat transformed into work taken off by the belt *is determined* usually by some form of absorption dyna-

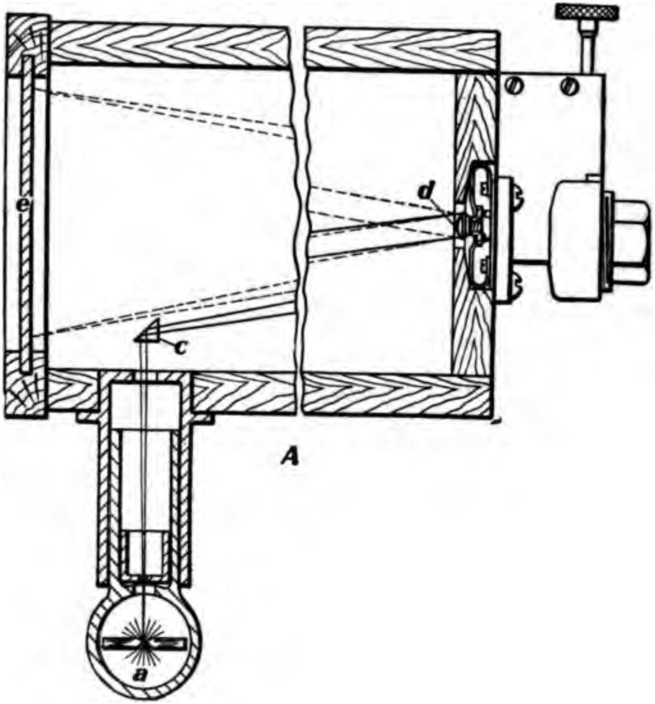
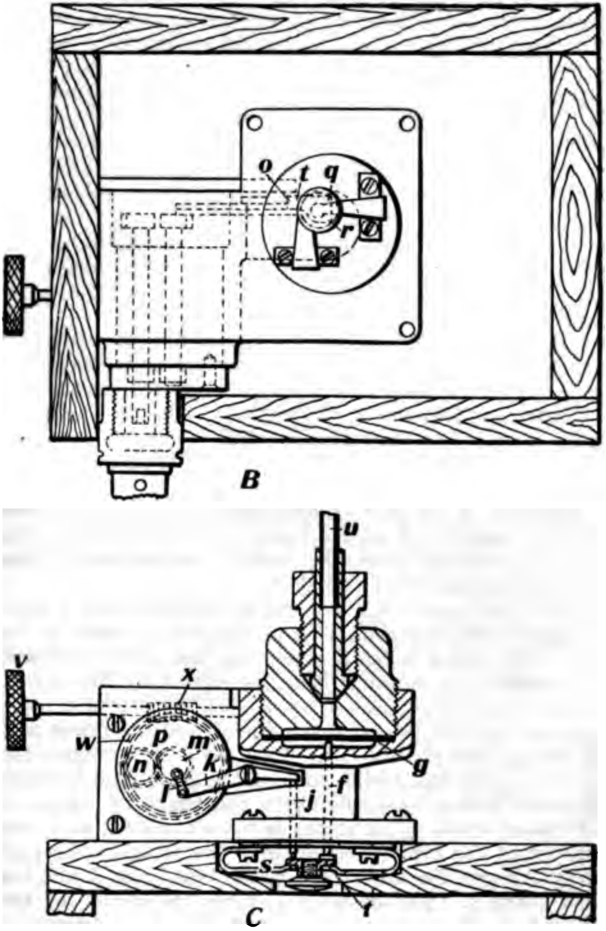


FIG. 79A.



FIGS. 79B AND C.

mometer, such as a prony brake. There are quite a number of ways in which such a brake may be made, the most common form being a series of wooden blocks which surround the flywheel and are tightened to the required extent by means of a screw. This brake has attached to it an arm, the end of which rests on supports on a platform scale, in order to measure the pull of the brake.

The difference between heat transformed into work upon the piston—indicated horsepower—and that given off at the belt—delivered horsepower—is the heat loss due to the friction of the engine.

The heat lost by the jacket water is measured by taking the temperature of the water as it enters and as it leaves the jacket and measuring the quantity of the water in pounds.

The heat carried off by the exhaust is measured by taking the temperature of the exhaust gases, by means of a pyrometer, and deducting from this the temperature of the mixture as it enters the engine. This temperature range (difference of temperature) multiplied by the weight of the gas and by its specific heat, determines the heat carried off by the exhaust.

The sum of the heat transformed into work, that lost in friction, carried off by the jacket water and lost by the exhaust, subtracted from the heat of combustion of the fuel, is that given off by radiation.

In Fig. 80 is shown a gas engine arranged for a complete test for the determination of the heat balance of the engine. The prony brake is shown applied to the flywheel, and the engine is running over, as shown by the arrow. The brake consists of the strap *a* with the blocks *b* bearing against the periphery of the wheel, and the brake arm made of the two boards *f*, *f*. Two cast- or wrought-iron angles *c c* are bolted to the ends of the band, and through them is the bolt *d* used to tighten the brake by means of the threaded crank *e*. A platform scale is placed at *k*, and the thrust of the brake is applied to the scale by the arm *f* acting through the knife-edge *g*, the iron block *h* and the wooden stand *j*. The distance *L* from the center of the crankshaft to the point where the knife-edge rests upon the block *h* is the length of the brake lever arm. As the pressure upon the scale is in a vertical direction, the measurement of the length of the brake arm should always be made horizontally or level.

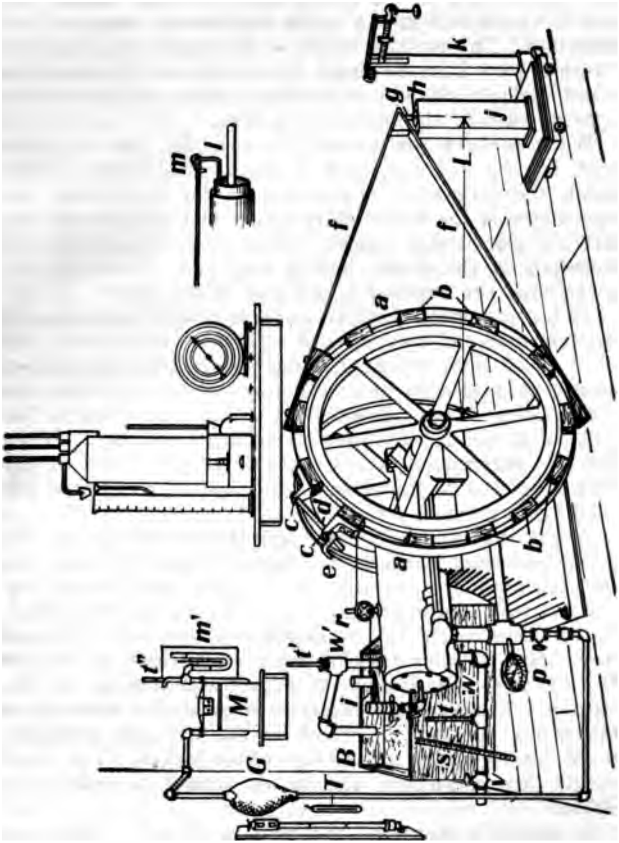


FIG. 80.

The indicator is shown at *i*, the cord from the indicator drum passing to the smaller drum of a reducing wheel *r*, and a cord from the larger wheel of the reduction device is attached to the pinion in some such manner as shown in the sketch at the upper right hand corner of the illustration. The method shown is by means of a piece of ⅜″ iron *l*, bent into the shape illustrated and flattened at the end opposite the eye *m* so that it may be fastened to the inside wall of the piston by screws.

Water enters the jacket through the pipe *w*, controlled by the valve *v*, and it leaves the water jacket through the pipe *w′*. Thermometers for measuring the temperature of the water as it enters and as it leaves the jacket are placed at *t* and *t′*. These thermometers do not set directly in the water, but in deep cups filled with oil, and the cups are exposed to the flow of the water.

In leaving the jacket the water is caught in the box *B* and either weighed or measured as is most convenient. In case it is desired to weigh the water, two receptacles should be provided, so that one may be emptied while the other is being filled. Attempting to catch the weight "on the fly" does not tend to give an accurate reading. When using two receptacles, the stream of water should be deflected from one to the other at the instant the signal for a reading is given.

When it is desired to measure the volume of the water, the tank may be made as follows in order to make the measurement easy to read. If the tank is 37 11-16″ by 37 11-16″, and the measuring stick *s* is graduated in half-inch divisions, each ½″ in depth will indicate 25 lb. of water. If a smaller tank is more convenient, it may be 26⅝″ by 26⅝″, and each 1″ depth will indicate 25 lb. of water. These measurements are given for water at a temperature of 150° F., which is the average temperature of water as it leaves the water-jacket. For very accurate determinations the water should of course be weighed.

A meter is shown on the wall at *M* and a gas bag, which should always be between the meter and the engine, at *g*. On the wall near the gas bag hangs the thermometer *T* for determining the temperature of the atmosphere and a barometer *X* for determining the atmospheric pressure. The temperature of the exhaust gases, just as they leave the engine is taken with the pyrometer *p*. The temperature of the gas as it enters the meter is given by the thermometer t'', and the pressure in inches of water by the manometer m′. In making a full report it is necessary to measure the volume of the air drawn into the engine. This is imperative in order to compute the volumetric efficiency of the engine and the heat lost by the exhaust. Computing the air drawn into the engine by assuming that it is the difference between the piston displacement and the volume of the gas per suction stroke, is simply wasting time. No gas engine ever made has a volumetric efficiency of 100%.

As the knowledge of just what an engine is doing and how it compares with other makes, is important to the manufacturer and the buyer, the method of making a complete test and the manner of working up the data will be explained fully. It is not expected that the complete thermal analysis, as outlined, will be made in every instance, and the reader is expected to select only such portions of the test as will suit his purpose. Before commencing the test, a form such as shown on page 290, and known as a log, should be made out. This log shows at the head of each column just what readings should be taken. The last four columns are for convenience in working up the data after the test, while the balance of the log is for data taken while the test is in progress.

In making the test, particularly when a large number of readings are to be taken, the engineer in charge should have a sufficient number of assistants to take the more important readings immediately after the signal. On the log, the readings that should be taken immediately upon the signal are those marked *, and in addition, the necessary indicator cards should be taken on the signal. For the complete series of readings therefore, the observers should be five in number.

Of these, one should man the brake to keep the pressure upon the scale at the same time point during one run. Thus, the brake will require the undivided attention of one man. A second man should measure the jacket water just at the signal or deflect the stream of water from one tank to the other when arranged for this method. A third man should read the meter or measure the fuel, catching the reading at the moment of the signal. The fourth man should read the air meter, while the fifth should take the indicator cards. The other readings should be taken as soon after the others as possible, as they should be divided among the operatives that do not take care of the brake. The reason for taking the certain readings just at the signal is in order that any error may be readily detected by comparing the reading of one period with those of another. If a continuous counter is used for taking the speed of the engine it may be advisable to read this at the signal. For runs of but an hour's duration, readings should be taken at intervals of five minutes, while for longer runs intervals of ten minutes will give satisfactory results.

The test should be divided into runs, each run being made at the same horsepower throughout. It is suggested that one run be made at the full power of the engine, all that it is capable of giving. Another run should be made at the rated horsepower load, and others at fractional loads, while for the final determination there should be no load and brake should be off. The run should be of sufficient duration for at least ten sets of readings, and a greater number will be found advantageous, if the test is a particularly important one.

The test should, if possible, be in charge of a competent engineer. He should be provided with a whistle, which should be blown as a preparatory signal thirty seconds before the time for taking the readings. Two blasts of the whistle should be the preparatory signal, and one blast should be the signal for taking the readings. Promptness should be observed, each man taking his post at the preparatory signal. No one but an observer engaged upon the test should be permitted to take a reading. Always enter the data upon the regular log and not upon loose slips of paper.

Either before or after the test, the diameter of the piston, the length of the stroke and the clearance should be accurately determined. The diameter of the piston should be taken when the engine is hot. To measure the clearance, set the crank on the dead center with the piston nearest the cylinder head. Weigh a quantity of water and from this water fill the compression space just full and no more, being careful not to spill any of the water. Then weigh the remaining water, and the difference will be the quantity necessary to fill the compression space. Correct for temperature in order to find the volume. Water at 39.1° F. weighs 62.425 lbs. per cu. ft., and to find the volne divide the weight by 62.425. If the water is much warmer than 39.1°, the weight may be found by the following formula:

$$\frac{62.425 \times 2}{\frac{t+461}{500}+\frac{500}{t+461}} = \text{Wt. per cu. ft.}$$

. Wherein t equals the temperature of the water.

This formula gives a close approximation to the correct weight. For the usual hydrant temperature, dividing by 62.425 will give a result sufficiently close for the purpose.

In making up a report of the test, the form on page 290 will be found convenient. The first line should be filled in with the name of the engine, the second with that of the manufacturer, the third line with the name of the party by whom the test is made or that of the engineering firm of which he is the representative, and the fourth line is for the locality at which the test is made, together with the date.

The greater number of the items in the report require no explanation, and only those will be explained that are likely to be misunderstood. The percentage of clearance is the volume of the clearance divided by the piston displacement. The ratio of the gas to the air is the volume of gas used during a certain period divided by the volume of air used during the same period. In those engines which draw air into the cylinder during a missed explosion, this ratio may be obtained only when the engine is running without missing an explosion. The temperature range for the jacket water is the difference between the temperature of the outlet and that of the inlet. The weight of the gas may be obtained from the gas company, by weighing the gas in a vacuum bottle or by computation from the analysis of the gas. If a balance is available it is a very simple matter to weigh a bottle of gas and compare that with the weight of the bottle filled with air. The volume of the bottle may be obtained with water in the same manner as the volume of the compression space.

One cubic foot of air weighs .080728 lb., 565.1 grains or 36.62 grams, at 32°F. Suppose the bottle is to contain *one pint of* air or gas. This would be 28.88 cubic inches or .01671 cu. ft. weighing 9.446 grains or 612.1 milligrams. Suppose the loss of weight was found to be 260 milligrams when the bottle was full of gas. Then the pint of gas would weigh 612.1—260 = 352.1 milligrams and a cu. ft. of the gas would weigh 352.1 ÷ .01671 = 21070 milligrams, or 21.07 grams per cu. ft. Reducing to pounds by multiplying by .002205 the weight of the gas is .04646 pounds per cu. ft.

For the convenience of those who may have the analysis of the gas, the following table gives the weights per cu. ft. of those gases which occur most frequently in the fuel gases, together with their specific heats at constant pressure:

Gas.	Lb. per cu. ft.	Spec. heat.
Marsh gas	.0447	.593
Olefines	.1174	.404
Hydrogen	.00559	3.409
Carbon monoxide	.0781	.248
Carbon dioxide	.1226	.217
Nitrogen	.0783	.244
Oxygen	.0892	.218
Water vapor	.0503	.48
Air (pure and dry)	.0807	.238

These weights are for the gases when at an atmospheric pressure of 14.7 lb. per sq. in. and a temperature of 32°F. The specific heat of the mixture may be found in the same manner as the weight when computing from the analysis, by multiplying the specific heat of each gas by the percentage of that gas in the mixture and adding the results. To find the weight of the mixture of air and gas, multiply the percentage of each by its weight per cu. ft. and this result, when added, will give the weight of a cu. ft. of the mixture.

When computing the results of the test, the volumes of the gases, both the air and the fuel should be reduced to that at 32°F and 30″ barometric pressure for the purpose of comparison. To make this reduction, the following formula should be used:

Let T = the temperature at the time of the test;
I = the pressure in inches of mercury;
v = *the volume* at this pressure and temperature;
V = *the volume* at 30″ of mercury and at 32° F.

Then for air,

$$V = \frac{vI \times 493}{30 \times (T - 461)}$$

This formula will also give a sufficiently close approximation when used for gas. In order to reduce the pressure in inches of water, as measured by the manometer, to inches of mercury, multiply by .0734. If the pressure of the gas is measured in lb. per sq. in. multiply by 2.033 to reduce to inches of mercury. These ratios are for 32°F., and will be sufficiently accurate for ordinary temperatures without temperature corrections.

The work in foot pounds per minute is found from the indicator diagram. To get this work, multiply together the M. E. P., the area of the cylinder in sq. in., the length of the stroke in feet and the number of explosions per minute. To find the heat supplied by the gas per hour multiply the number of cu. ft. used per hour by the heat value of the gas per cu. ft. The heat absorbed by the jacket water is the weight of the water passed through the jacket multiplied by the temperature range. The heat absorbed in work is the number of ft. lb. per hour divided by 778. The figure 778 is the mechanical equivalent of heat, that is the number of ft. lb. equivalent to one B. T. U. The latter quantity is the British Thermal Unit, and is the quantity of heat necessary to raise one pound of water one degree of the Fahrenheit scale, in temperature.

The specific heat of gases varies with rise of temperature, and for this reason the heat carried off by the exhaust gases can be found only approximately.

The formula for this determination is as follows:

Let S = the specific heat of the mixture;
w = the weight of one cu. ft. of the mixture in lb.
q = the quantity in cu. ft. of the mixture exhausted per hour;
T = the temperature of the room, or the average temperature of the air and gas entering the engine;
T' = temperature of exhaust by pyrometer;
U = quantity of heat in B. T. U. carried off by exhaust in one hour.

Then $U = Swq\ (T' - T)$.

The volume of the burned mixture passing through the exhaust is the sum of all the gas and the air used by the engine, and it should include all air passing from the exhaust for any reason.

Since the specific heat per cu. ft. of the majority of the gases exhausted is .0191 for a rough approximation the following formula will give the value very closely:

$U = .0191q\ (T' - T)$.

The remaining heat unaccounted for after finding the heat transformed into work, given off through the jacket water and lost by the exhaust, is credited to loss by radiation. For the greatest refinement, the exhaust gas should be analyzed and the quantity of unburned fuel determined. This forms a fifth item of heat loss, and should be deducted in advance of computing the radiation.

The thermal efficiency of the engine is the quotient of the heat absorbed in work, divided by the heat supplied by the fuel. It is usually written as percentage. There are two figures for thermal efficiency, one based on the indicated and the other on the delivered horsepower.

The indicated horsepower is computed from the indicator diagrams. The area of the diagram is found by means of a planimeter*, and this area, divided by the length of the diagram in inches, gives the mean ordinate. The M. E. P. (mean effective pressure) is found by multiplying the mean ordinate by the scale of the indicator spring. When the indicator is fitted with special ¼" area piston, the scale of the spring should be multiplied by two, as the springs are usually marked for ½" area piston.

To find the I. H. P. use the following formula:

$$\text{I.H.P.} = \frac{PLAN}{33{,}000}$$

Wherein P = the M. E. P.;
L = the length of the stroke in feet;
A = the area of the piston in sq. in.;
N = the number of *explosions* per minute.

To find the D. H. P., the length of the brake arm l, (Fig. 80) the r. p. m. and the pressure on the scale must be known. The formula is:

$D.\ H.\ P. = .0001904\ pln.$

Wherein p = net pressure in pounds on the scale;
l = length of brake arm in feet;
n = r. p. m.

---

*For the method of using the planimeter the reader is referred to works on this instrument, as space will not permit of its treatment in this volume.

Included in the pressure shown on the scale there is usually a small pressure exerted by the weight of the brake arm itself, and perhaps some other unbalanced portion of the brake. To find this pressure (the tare) proceed as follows. When the engine is not running a man should stand upon the scale platform and the scale balanced. Note his weight. Then balance the scale while he has hold of the end of the brake arm where it rests on the scale and is pushing slowly downward. Balance again while he is pulling the arm slowly upward. Be sure the arm is in motion when the scale is balanced. Add together the weights found while moving the arm, divide the sum by two and subtract the weight of the man. The result will be the pressure due to the unbalanced portion of the arm. This pressure should be subtracted from the pressure recorded on the scale to obtain the net or actual pressure exerted by the friction of the brake. Thus, suppose the weight of the man is 160 lb., and that, when the arm is being pulled up, the scale balance at 180 lb. and when the arm is being pushed down the scale balances at 170 lb. Then the effect of the arm would be

$$\frac{170 + 180}{2} - 160 = 15 \text{ lb.}$$

To the tare should be added the weight of the block j, Fig. 80.

Log of Test.

....Gas Engine. Made at..

....At.., 19...

				Temp. Degrees Fahr.						Dynamometer			Fuel		Air			Calorimeter							
							Jacket Water																		
Number	Time	r.p.m.	Explosions per min.	Room	Exhaust gas	Fuel gas	Inlet	Outlet	Pounds* or cu. ft.	Scale reading	Net press. lb.	Ft. lb. per rev.	Gas cu. ft. or* liquid fuel lb.	Pres. inches water	Cu. ft.*	Press. barom.	Press. at 32° Fahr.	Temp. inlet	Temp. outlet	Cu. cm. water	Condensed water	B.T.U. low value.	M.E.P.	I.H.P.	D.H.P.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
1																									
2																									
3																									
4																									
5																									

REPORT OF TEST.

.... *Engine*

Built by ....*Company*

Test made by...

At ....19..

Diameter of piston..........in. ....

DIMENSIONS OF ENGINE.		First Cylin.	Second Cylin.
Diameter of piston	in.	....	....
Diameter of piston-rod	in.	....	....
Area of piston	sq. in.	....	....
Piston displacement	cu. ft.	....	....
Clearance	cu. ft.	....	....
Clearance	per cent.	....	....

DATA.

Duration of trial		....
Gas per hour	cu. ft.	....
Gasoline or other liquid fuel per hour	lb.	....
Air per hour	cu. ft.	....
Mixture per hour	cu. ft.	....
Ratio gas to air		....
Jacket water per hour	lb.	....
Jacket water temperature, inlet, average	F°	....
Jacket water temperature, outlet, average	F°	....
Jacket water, temperature range	F°	....
Revolutions per minute, average		....
Revolutions per hour		....
Explosions per minute, average		....
Explosions per hour		....
Speed, no load	r. p. m	....
Speed, full load	r. p. m.	....
Speed range	r. p. m.	....
Speed range	per cent.	....
Temperature exhaust	F°	....
Temperature room	F°	....
Length of brake arm	ft.	....
Brake load, average	lb.	....

Gas, weight of cu. ft.....................lb. ....
Air, weight of cu. ft.....................lb. ....
Mixture, weight of cu. ft..............lb. ....
Specific heat, air.......................... ....
Specific heat, gas.......................... ....
Specific heat, mixture ....
Heat value per cu. ft. gas (low value)....... ....

RESULTS.

Work, ft. lb. per min., average.............. ....
Work, ft. lb. per hour, average............ ....
D. H. P., average ....
Indicated M. E. P., average ....lb. ....
I. H. P., average.......................... ....
Gas per I. H. P. per hour...............cu. ft. ....
Gas per D. H. P. per hour ....cu. ft. ....
Gasoline (liquid fuel) per I. H. P. per hr..lb. ....
Gasoline (liquid fuel) per D. H. P. per hr..lb. ....
Mechanical efficiency (D. H. P. ÷ I. H. P.)
per cent. ....
Friction loss (I. H. P. — D. H. P.)......H. P. ....

HEAT PER HOUR.

Supplied by gas...................B. T. U. ....
Absorbed by jacket water..........B. T. U. ....
Exhausted ....B. T. U. ....
Absorbed in work..................B. T. U. ....
Radiated ....B. T. U. ....

EFFICIENCIES.

Thermal efficiency based on I. H. P...per cent. ....
Thermal efficiency based on D. H. P..per cent. ....
B. T. U. per I. H. P....................... ....
B. T. U. per D. H. P...................... ....
Volumetric efficiency ....per cent.. ....

CHAPTER XXXVII.

SELECTION

In the selection of a gas engine, be sure you have a definite idea of just what machinery it is to drive and how much future installation you will expect the engine to take care of. Find out as nearly as possible just what power is required to drive this machinery. This information may be obtained, usually, from the makers of the machines. Then allow for the line shafting at least 65 per cent as much power as is required to drive the machinery. Thus, if 100 H. P. is required to drive the machinery, allow 65 H. P. additional to drive the shafting or 165 total H. P. And in addition to the horse power of the engine, be sure to specify the class of machinery the engine is to drive, and select an engine with sufficiently close regulation for this machinery. For example, an engine with sufficiently close regulation for driving machine tools may not be suited to driving alternators in parallel, and an engine with sufficiently close regulation for the latter duty is generally higher in first cost than is necessary for driving machine tools.

Quite often the engine salesman has had sufficient experience to advise you all along these lines, but remember that it is his business to get sales, and that occasionally a salesman is apt to overrate the capabilities of an engine in order to make a sale. It is a good plan, before deciding on a certain make of engine, to talk to several users of gas engines who are employing that make of engine for driving the same class of machinery that you intend to drive, and using about the same horsepower that you will require.

Investigate especially how much the engine has cost the owner for repairs, and how often and for how long the engine has been laid up for repairs. Learn how long the engine has been in use, and whether it has been in constant service since its installation. Make it a point to see the engine in operation, and watch it while running. If the engine is in a filthy condition, or poorly installed, or if it is running with its parts out of adjustment when they could easily be put in good running order, score a few points in favor of the engine and against the engineer. An engine that will do fairly good work under bad management has something to recommend it. If, on the other hand, the engine is clean, and has all possible adjustments taken care of, but is running in a noisy, jerky way, score against the engine. If the engine is overloaded, and carries its fullest capacity for long periods at a time without distress or heating of the cylinder and bearings, it is undoubtedly good in this respect. On the other hand, an engine that gives trouble from overheating may not be at fault in itself, but be supplied with a poor quality of oil, or insufficient oil, or it may be running on too weak a mixture.

If an engine is counterbalanced with the counterweights in the flywheel, stand in line with the axis of the cylinder and observe if the flywheels are running true at the side. If a flywheel is observed to sway back and forth near the rim, it is a sign that the shaft is bent or that the wheel is either loose or not mounted true. Usually the cause is a weak shaft that has bent under the strain. Find out if the bearings give much trouble from heating, especially when the engine is working under full load. Determine if frequent renewal of parts has been found necessary, especially in the igniter and the bearings. The occasional renewal of igniter points is to be expected as well as a certain amount of wear, but frequent repairs is not a good point in favor of any engine. Another important feature is the interchangeability of parts. Will a part ordered from the factory need to be fitted to your engine, or will it fit into place just as received from the factory? This is especially important in small engines, such as marine and automobile motors, for which the parts should be interchangeable.

Find out how much attention from the operator the engine requires in a day, and just what the nature of this attention is. If the engine is required to drive a dynamo, watch the lights for fluctuation while the engine is running. The best way to observe the light is to do so indirectly, as, for example, reading by the light. If there is no perceptible winking, the engine may be considered as giving good regulation for this service. If the pressure on the line is normal at 100 volts, the variation as shown by the voltometer should not exceed two volts, or on a fifty volt circuit, one volt.

The fuel consumption of an engine is a very important item, and increases in importance as the engine increases in power. At full load the fuel consumption for a modern gas engine of 100 H. P. and over should not exceed eleven cu. ft. of natural gas per D. H. P. hour. This is based on a gas having a heat value of at least 1,000 B. T. U. per cu. ft. For gasoline the fuel consumption should be approximately one pint per B. H. P. hour, at full load. Where you can secure it, it is a good plan to have a guarantee of fuel consumption when the engine is purchased. The guarantee should cover full load, three-quarter load and half load. The engine will give its best economy at full load, and some manufacturers are not willing to give a guarantee of fuel economy on any but full load.

In the selection of an engine for any purpose, take that one, in preference, that has the fewest number of parts. Buy an engine that has the parts accessible and so arranged that the least possible amount of dismantling is required to get at any part for repairs, or to remove it for replacement. Do not be deceived by the fact that the parts are encased, for an engine may be just as complicated when the parts are covered up as when they are exposed to view, and the parts are usually much harder to get at. At the same time, the engine with enclosed parts may be an excellent one, and it is certainly better for the parts to be enclosed when the engine has to be used in dusty places, as in farm tractor work. In this instance, it is imperative that all the working parts of the engine be enclosed, on account of the flying dirt to which the engine is exposed, especially when plowing. With reference to the complication or accessibility of the engine, the author wishes to impress the reader with two things, one is that the simpler the engine and the fewer the number of parts, the less likely is the engine to get out of order, and the other is that it should not be necessary to take the engine entirely to pieces to repair some trifling piece or to make some small adjustment.

WROUGHT IRON WELDED STEAM, GAS AND WATER PIPE.

DIAMETER			Thickness	TRANSVERSE AREAS		Nominal Weight per foot	Number of Threads per inch of Screw	Tap Drill
Nominal Internal	Actual External	Actual Internal		External	Internal			
Inches	Inches	Inches	Inches	Sq. Ins.	Sq. Ins.	Pounds		
⅛	.405	.27	.068	.129	.0573	.244	27	11/32
¼	.54	.364	.088	.229	.1041	.425	18	15/32
⅜	.675	.494	.091	.358	.1917	.568	18	19/32
½	.84	.623	.109	.554	.3048	.852	14	¾
¾	1.05	.824	.113	.866	.5333	1.130	14	31/32
1	1.315	1.048	.134	1.358	.8626	1.678	11½	1 7/32
1¼	1.66	1.38	.14	2.164	1.496	2.272	11½	1 17/32
1½	1.9	1.611	.145	2.835	2.038	2.717	11½	1 25/32
2	2.375	2.067	.154	4.43	3.356	3.652	11½	2 7/32
2½	2.875	2.468	.204	6.492	4.784	5.793	8	2 11/16
3	3.5	3.067	.217	9.621	7.388	7.575	8	3 5/16
3½	4.	3.548	.226	12.566	9.887	9.202	8	3 13/16
4	4.5	4.026	.237	15.904	12.73	10.889	8	4 5/16
4½	5.	4.508	.246	19.635	15.961	12.642	8	4 13/16
5	5.563	5.045	.259	24.306	19.99	14.617	8	5 11/32
6	6.625	6.065	.28	34.472	28.888	18.974	8	6 11/32
7	7.625	7.023	.301	45.664	38.738	23.544	8	7 7/16
8	8.625	7.982	.322	58.426	50.04	28.809	8	8 7/16

CIRCUMFERENCES AND AREAS OF CIRCLES

ADVANCING BY EIGHTHS.

Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	.04909	.00019	1.	3.1416	.7854
1/32	.09818	.00077	1/16	3.3379	.8866
3/64	.14726	.00173	⅛	3.5343	.9940
1/16	.19635	.00307	3/16	3.7306	1.1075
3/32	.29452	.00690	¼	3.9270	1.2272
⅛	.39270	.01227	5/16	4.1233	1.3530
5/32	.49087	.01917	⅜	4.3197	1.4849
3/16	.58905	.02761	7/16	4.5160	1.6230
7/32	.68722	.03758	½	4.7124	1.7671
			9/16	4.9087	1.9175
¼	.78540	.04909	⅝	5.1051	2.0739
9/32	.88357	.06213	11/16	5.3014	2.2365
5/16	.98175	.07670	¾	5.4978	2.4053
11/32	1.0799	.09281	13/16	5.6941	2.5802
⅜	1.1781	.11045	⅞	5.8905	2.7612
13/32	1.2763	.12962	15/16	6.0868	2.9483
7/16	1.3744	.15033			
15/32	1.4726	.17257	2.	6.2832	3.1416
			1/16	6.4795	3.3410
½	1.5708	.19635	⅛	6.6759	3.5466
17/32	1.6690	.22166	3/16	6.8722	3.7583
9/16	1.7671	.24850	¼	7.0686	3.9761
19/32	1.8653	.27688	5/16	7.2649	4.2000
⅝	1.9635	.30680	⅜	7.4613	4.4301
21/32	2.0617	.33824	7/16	7.6576	4.6664
11/16	2.1598	.37122	½	7.8540	4.9087
23/32	2.2580	.40574	9/16	8.0503	5.1572
			⅝	8.2467	5.4119
¾	2.3562	.44179	11/16	8.4430	5.6727
25/32	2.4544	.47937	¾	8.6394	5.9396
13/16	2.5525	.51849	13/16	8.8357	6.2126
27/32	2.6507	.55914	⅞	9.0321	6.4918
⅞	2.7489	.60132	15/16	9.2284	6.7771
29/32	2.8471	.64504			
15/16	2.9452	.69029	3.	9.4248	7.0686
31/32	3.0434	.73708	1/16	9.6211	7.3662

CIRCUMFERENCES AND AREAS—*Continued*

Diam.	Circum.	Area	Diam.	Circum.	Area
3. ⅛	9.8175	7.6699	**5.** ¾	18.064	25.967
3/16	10.014	7.9798	13/16	18.261	26.535
¼	10.210	8.2958	⅞	18.457	27.109
5/16	10.407	8.6179	15/16	18.653	27.688
⅜	10.603	8.9462			
7/16	10.799	9.2806	**6.**	18.850	28.274
½	10.996	9.6211	⅛	19.242	29.465
9/16	11.192	9.9678	¼	19.635	30.680
⅝	11.388	10.321	⅜	20.028	31.919
11/16	11.585	10.680	½	20.420	33.183
¾	11.781	11.045	⅝	20.813	34.472
13/16	11.977	11.416	¾	21.206	35.785
⅞	12.174	11.793	⅞	21.598	37.122
15/16	12.370	12.177			
			7.	21.991	38.485
4.	12.566	12.566	⅛	22.384	39.871
1/16	12.763	12.962	¼	22.776	41.282
⅛	12.959	13.364	⅜	23.169	42.718
3/16	13.155	13.772	½	23.562	44.179
¼	13.352	14.186	⅝	23.955	45.664
5/16	13.548	14.607	¾	24.347	47.173
⅜	13.744	15.033	⅞	24.740	48.707
7/16	13.941	15.466			
½	14.137	15.904	**8.**	25.133	50.265
9/16	14.334	16.349	⅛	25.525	51.849
⅝	14.530	16.800	¼	25.918	53.456
11/16	14.726	17.257	⅜	26.311	55.088
¾	14.923	17.721	½	26.704	56.745
13/16	15.119	18.190	⅝	27.096	58.426
⅞	15.315	18.665	¾	27.489	60.132
15/16	15.512	19.147	⅞	27.882	61.862
5.	15.708	19.635	**9.**	28.274	63.617
1/16	15.904	20.129	⅛	28.667	65.397
⅛	16.101	20.629	¼	29.060	67.201
3/16	16.297	21.135	⅜	29.452	69.029
¼	16.493	21.648	½	29.845	70.882
5/16	16.690	22.166	⅝	30.238	72.760
⅜	16.886	22.691	¾	30.631	74.662
7/16	17.082	23.221	⅞	31.023	76.589
½	17.279	23.758			
9/16	17.475	24.301	**10.**	31.416	78.540
⅝	17.671	24.850	⅛	31.809	80.516
11/16	17.868	25.406	¼	32.201	82.516

CIRCUMFERENCES AND AREAS—*Continued.*

Diam.	Circum.	Area	Diam.	Circum.	Area
10.⅜	32.594	84.541	**15.¼**	47.909	182.65
½	32.987	86.590	⅜	48.302	185.66
⅝	33.379	88.664	½	48.695	188.69
¾	33.772	90.763	⅝	49.087	191.75
⅞	34.165	92.886	¾	49.480	194.83
11.	34.558	95.033	⅞	49.873	197.93
⅛	34.950	97.205	**16.**	50.265	201.06
¼	35.343	99.402	⅛	50.658	204.22
⅜	35.736	101.62	¼	51.051	207.39
½	36.128	103.87	⅜	51.444	210.60
⅝	36.521	106.14	½	51.836	213.82
¾	36.914	108.43	⅝	52.229	217.08
⅞	37.306	110.75	¾	52.622	220.35
12.	37.699	113.10	⅞	53.014	223.65
⅛	38.092	115.47	**17.**	53.407	226.98
¼	38.485	117.86	⅛	53.800	230.33
⅜	38.877	120.28	¼	54.192	233.71
½	39.270	122.72	⅜	54.585	237.10
⅝	39.663	125.19	½	54.978	240.53
¾	40.055	127.68	⅝	55.371	243.98
⅞	40.448	130.19	¾	55.763	247.45
13.	40.841	132.73	⅞	56.156	250.95
⅛	41.233	135.30	**18.**	56.549	254.47
¼	41.626	137.89	⅛	56.941	258.02
⅜	42.019	140.50	¼	57.334	261.59
½	42.412	143.14	⅜	57.727	265.18
⅝	42.804	145.80	½	58.119	268.80
¾	43.197	148.49	⅝	58.512	272.45
⅞	43.590	151.20	¾	58.905	276.12
14.	43.982	153.94	⅞	59.298	279.81
⅛	44.375	156.70	**19.**	59.690	283.53
¼	44.768	159.48	⅛	60.083	287.27
⅜	45.160	162.30	¼	60.476	291.04
½	45.553	165.13	⅜	60.868	294.83
⅝	45.946	167.99	½	61.261	298.65
¾	46.338	170.87	⅝	61.654	302.49
⅞	46.731	173.78	¾	62.046	306.35
15.	47.124	176.71	⅞	62.439	310.24
⅛	47.517	179.67	**20.**	62.832	314.16

CIRCUMFERENCES AND AREAS—*Continued.*

Diam.	Circum.	Area	Diam.	Circum.	Area
20.⅛	63.225	318.10	**25.**	78.540	490.87
¼	63.617	322.06	⅛	78.933	495.79
⅜	64.010	326.05	¼	79.325	500.74
½	64.403	330.06	⅜	79.718	505.71
⅝	64.795	334.10	½	80.111	510.71
¾	65.188	338.16	⅝	80.503	515.72
⅞	65.581	342.25	¾	80.896	520.77
21.	65.973	346.36	⅞	81.289	525.84
⅛	66.366	350.50	**26.**	81.681	530.93
¼	66.759	354.66	⅛	82.074	536.05
⅜	67.152	358.84	¼	82.467	541.19
½	67.544	363.05	⅜	82.860	546.35
⅝	67.937	367.28	½	83.252	551.55
¾	68.330	371.54	⅝	83.645	556.76
⅞	68.722	375.83	¾	84.038	562.00
22.	69.115	380.13	⅞	84.430	567.27
⅛	69.508	384.46	**27.**	84.823	572.56
¼	69.900	388.82	⅛	85.216	577.87
⅜	70.293	393.20	¼	85.608	583.21
½	70.686	397.61	⅜	86.001	588.57
⅝	71.079	402.04	½	86.394	593.96
¾	71.471	406.49	⅝	86.786	599.37
⅞	71.864	410.97	¾	87.179	604.81
23.	72.257	415.48	⅞	87.572	610.27
⅛	72.649	420.00	**28.**	87.965	615.75
¼	73.042	424.56	⅛	88.357	621.26
⅜	73.435	429.13	¼	88.750	626.80
½	73.827	433.74	⅜	89.143	632.36
⅝	74.220	438.36	½	89.535	637.94
¾	74.613	443.01	⅝	89.928	643.55
⅞	75.006	447.69	¾	90.321	649.18
24.	75.398	452.39	⅞	90.713	654.84
⅛	75.791	457.11	**29.**	91.106	660.52
¼	76.184	461.86	⅛	91.499	666.23
⅜	76.576	466.64	¼	91.892	671.96
½	76.969	471.44	⅜	92.284	677.71
⅝	77.362	476.26	½	92.677	683.49
¾	77.754	481.11	⅝	93.070	689.30
⅞	78.147	485.98	¾	93.462	695.13
			⅞	93.855	700.98

CIRCUMFERENCES AND AREAS—*Continued.*

Diam.	Circum.	Area	Diam.	Circum.	Area
30.	94.248	706.86	**35.**	109.956	962.11
⅛	94.640	712.76	⅛	110.348	969.00
¼	95.033	718.69	¼	110.741	975.91
⅜	95.426	724.64	⅜	111.134	982.84
½	95.819	730.62	½	111.527	989.80
⅝	96.211	736.62	⅝	111.919	996.78
¾	96.604	742.64	¾	112.312	1003.8
⅞	96.997	748.69	⅞	112.705	1010.8
31.	97.389	754.77	**36.**	113.097	1017.9
⅛	97.782	760.87	⅛	113.490	1025.0
¼	98.175	766.99	¼	113.883	1032.1
⅜	98.567	773.14	⅜	114.275	1039.2
½	98.960	779.31	½	114.668	1046.3
⅝	99.353	785.51	⅝	115.061	1053.5
¾	99.746	791.73	¾	115.454	1060.7
⅞	100.138	797.98	⅞	115.846	1068.0
32.	100.531	804.25	**37.**	116.239	1075.2
⅛	100.924	810.54	⅛	116.632	1082.5
¼	101.316	816.86	¼	117.024	1089.8
⅜	101.709	823.21	⅜	117.417	1097.1
½	102.102	829.58	½	117.810	1104.5
⅝	102.494	835.97	⅝	118.202	1111.8
¾	102.887	842.39	¾	118.596	1119.2
⅞	103.280	848.83	⅞	118.988	1126.7
33.	103.673	855.30	**38.**	119.381	1134.1
⅛	104.065	861.79	⅛	119.773	1141.0
¼	104.458	868.31	¼	120.166	1149.1
⅜	104.851	874.85	⅜	120.559	1156.6
½	105.243	881.41	½	120.951	1164.2
⅝	105.636	888.00	⅝	121.344	1171.7
¾	106.029	894.62	¾	121.737	1179.3
⅞	106.421	901.26	⅞	122.129	1186.9
34.	106.814	907.92	**39.**	122.522	1194.6
⅛	107.207	914.61	⅛	122.915	1202.3
¼	107.600	921.32	¼	123.308	1210.0
⅜	107.992	928.06	⅜	123.700	1217.7
½	108.385	934.82	½	124.093	1225.4
⅝	108.778	941.61	⅝	124.486	1233.2
¾	109.170	948.42	¾	124.878	1241.0
⅞	109.563	955.25	⅞	125.271	1248.8

CIRCUMFERENCES AND AREAS—*Continued.*

Diam.	Circum.	Area	Diam.	Circum.	Area
40.	125.664	1256.6	**45.**	141.372	1590.4
⅛	126.056	1264.5	⅛	141.764	1599.3
¼	126.449	1272.4	¼	142.157	1608.2
⅜	126.842	1280.3	⅜	142.550	1617.0
½	127.235	1288.2	½	142.942	1626.0
⅝	127.627	1296.2	⅝	143.335	1634.9
¾	128.020	1304.2	¾	143.728	1643.9
⅞	128.413	1312.2	⅞	144.121	1652.9
41.	128.805	1320.3	**46.**	144.513	1661.9
⅛	129.198	1328.3	⅛	144.906	1670.9
¼	129.591	1336.4	¼	145.299	1680.0
⅜	129.983	1344.5	⅜	145.691	1689.1
½	130.376	1352.7	½	146.084	1698.2
⅝	130.769	1360.8	⅝	146.477	1707.4
¾	131.161	1369.0	¾	146.869	1716.5
⅞	131.554	1377.2	⅞	147.262	1725.7
42.	131.947	1385.4	**47.**	147.655	1734.9
⅛	132.340	1393.7	⅛	148.048	1744.2
¼	132.732	1402.0	¼	148.440	1753.5
⅜	133.125	1410.3	⅜	148.833	1762.7
½	133.518	1418.6	½	149.226	1772.1
⅝	133.910	1427.0	⅝	149.618	1781.4
¾	134.303	1435.4	¾	150.011	1790.8
⅞	134.696	1443.8	⅞	150.404	1800.1
43.	135.088	1452.2	**48.**	150.796	1809.6
⅛	135.481	1460.7	⅛	151.189	1819.0
¼	135.874	1469.1	¼	151.582	1828.5
⅜	136.267	1477.6	⅜	151.975	1837.9
½	136.659	1486.2	½	152.367	1847.5
⅝	137.052	1494.7	⅝	152.760	1857.0
¾	137.445	1503.3	¾	153.153	1866.5
⅞	137.837	1511.9	⅞	153.545	1876.1
44.	138.230	1520.5	**49.**	153.938	1885.7
⅛	138.623	1529.2	⅛	154.331	1895.4
¼	139.015	1537.9	¼	154.723	1905.0
⅜	139.408	1546.6	⅜	155.116	1914.7
½	139.801	1555.3	½	155.509	1924.4
⅝	140.194	1564.0	⅝	155.902	1934.2
¾	140.586	1572.8	¾	156.294	1943.9
⅞	140.979	1581.6	⅞	156.687	1953.7

STRENGTH OF MATERIALS.

Material	Tensile Lbs. per sq. in.	Elastic Limit Lbs. per sq. in.
Aluminum, Cast	15,000	6,500
Aluminum, Sheet	24,000	12,000
Aluminum, Bars	28,000	14,000
Aluminum, Wire	30,000 to 65,000	16,000 to 30,000
Aluminum 92% Copper 8%	19,000	
Ditto, Rolled	50,000	
Brass (Copper, Tin, Zinc)	32,000 Ave.	
Bronze (Copper, Tin)	28,000 Ave.	16,000 Ave.
Bronze, Manganese	60,000	30,000
Iron, Castings, Common	21,000	6,000
Iron, Cylinders	24,000 to 28,000	
Iron, Malleable	41,000	19,000
Iron, Wrought	48,000	26,000
Magnalium, Cast	20,000	
Magnalium, Rolled	40,000	
Steel Castings	56,000 to 80,000	
Steel, Forged 0.20% C	55,000	28,000
" " 0.30% C	75,000	37,000
" " 0.40% C	85,000	43,000
" " 0.50% C	95,000	48,000
Steel with 3.5% Nickel and Carbon = 0.20%	85,000	48,000
Ditto 0.30%	95,000	60,000
Ditto 0.40%	110,000	72,000
Ditto 0.50%	125,000	85,000
Steel, Chrome-Vanadium	127,800	110,100

SPECIFIC GRAVITY OF METALS.

Metal	Spec. Gravity	Wt. per Cu. In.
Aluminum, Pure, Cast	2.67	.0963
Aluminum 92%, Copper 8%	2.83	.1022
Brass, Copper and Zinc	8.20 Ave.	.2959
Bronze, Copper and Tin	8.853	.3195
Copper	8.853	.3195
Iron, Cast	7.218	.2604
Iron, Wrought	7.70	.2779
Lead	11.38	.4106
Manganese	8.	.2887
Magnesium	1.75	.0632
Magnalium	2.4	.0866
Mercury	13.58	.4900
Nickel	8.8	.3175
Steel	7.854	.2834
Tin	7.35	.2652
Zinc	7.00	.2526

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